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3. REPORT TITLE

THE EFFECT OF THE FORM OF AN ACTIVE GAS JET ON THE CHARACTERISTICS OF A GAS EJECTOR WITH SHORT MIXING CHAMBER

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Translation

5. AUTHOR(S) (First name, middle initial, last name)

Bordovitsyn, Yu. A.; Koval'nogov, A. F.; Filin, V. A.

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13. ABSTRACT

This article examines ejectors with short mixing chambers because of their improved technical characteristics. An experimental model of this type of ejector was built and investigated. It was found that the use of gas nozzles with short ejector mixing chambers reduces the noise level of the emergent stream and increases the air ejection coefficient by 1.25-3.75 times with a change in a given hydraulic resistance from 0.94 to 1.03.

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14. KEY WORDS	LINK A		LINK B		LINK C	
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Gas Ejector Gas Jet						

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1. ORIGINATING ACTIVITY (Corporate author) Foreign Technology Division Air Force Systems Command U. S. Air Force	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED
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3. REPORT TITLE

ON THE CALCULATION OF THE MAXIMUM DEGREE OF COMPRESSION OF AN EJECTOR

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

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5. AUTHOR(S) (First name, middle initial, last name)

Kalmykov, I. I. and Mosin, I. I.

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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Foreign Technology Division Wright-Patterson AFB, Ohio

13. ABSTRACT

This article discusses methods for determining the maximum degree of ejector compression. The inadequacies of methods based on the one-dimensional theory of a gas flow are pointed out and suggestions for improvements are made.

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THE RATIONAL METHOD OF SELECTING PARAMETERS AND CALCULATING SINGLE-STAGE GAS TURBINES

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Strunkin, V. A.

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13. ABSTRACT

This article attempts to define a method for the efficient selection of parameters and design factors of single-stage turbines. The suggested procedure overcomes many of the shortcomings of previously used methods.

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14. KEY WORDS	LINK A		LINK B		LINK C	
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Foreign Technology Division
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A GENERALIZATION OF THE RESULTS OF MEASUREMENTS OF THE FINENESS OF FUEL ATOMIZATION OF MECHANICAL AND AIR-MECHANICAL INJECTORS OF THE PRESSURE-JET TYPE

DESCRIPTIVE NOTES (Type of report and inclusive dates)

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Dyatlov, I. N.

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12. SPONSORING MILITARY ACTIVITY

Foreign Technology Division
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13. ABSTRACT

This article presents the results of computer processing of experimental analyses to measure the degree of atomization of mechanical and air-mechanical pressure-jet axifugal injectors.

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3. REPORT TITLE
ON THE CALCULATION OF TEMPERATURE FIELD IN A COOLED TURBINE BLADE WITH LONGITUDINAL COOLING CHANNELS

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)
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5. AUTHOR(S) (First name, middle initial, last name)
Lokay, V. I. and Sharapov, A. V.

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	Foreign Technology Division Wright-Patterson AFB, Ohio

13. ABSTRACT

This article studies the problems of and solutions to the determination of temperatures inside cooled turbine blades having lengthwise cooling passages. Various methods and calculations are given.

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3. REPORT TITLE

TURBULENCE IN THE ZONE OF INTERPLAY OF JETSTREAMS WITH THE FLOW

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

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5. AUTHOR(S) (First name, middle initial, last name)

Kosterii, V. A.; Dudin, La. A.; Rogozhin, Alekseyev, Yu. S.; and Shalayev, G. M.

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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	Foreign Technology Division Wright-Patterson AFB, Ohio

13. ABSTRACT

This article discusses the mechanism of turbulence of the flow in the jetstream interaction zone. Known methods for determining turbulence characteristics are mentioned and new approaches to this problem are suggested.

14. KEY WORDS	LINK A		LINK B		LINK C	
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Gas Flow Jet Stream Turbulent Flow						

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АВИАЦИОННЫЙ ИНСТИТУТ

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Авиационные двигатели

КАЗАНЬ
1969

TABLE OF CONTENTS

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Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* <u>ye</u> initially, after vowels, and after ъ, ь; <u>e</u> elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

Translator's note: On several occasions, symbols found in formulae and calculations appear to have been rendered incorrectly in the original document. They will be shown exactly as they appear in the original.

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THE EFFECT OF THE FORM OF AN ACTIVE GAS JET ON THE CHARACTERISTICS OF A GAS EJECTOR WITH SHORT MIXING CHAMBER

Yu. A. Bordovitsyn, A. F. Koval'nogov, and V. A. Filin

In the gas passages of gas-turbine devices and in a number of other gas-jet devices widespread application is being given to ejectors with short mixing chambers (0.5-2 bore). Shortening of the mixing chamber is accompanied by a decrease in weight and the overall size of ejector equipment.

Improvement of the characteristics of such ejectors - an increase in the coefficient of ejection n, a decrease in hydraulic resistance Δp, the level of the air noise of the jet emergent from the ejector L, - is of significant practical interest.

For an appraisal of the effect of the geometric form of the jet of an active gas on the cited characteristics of a gas ejector experimental analysis on an ejector model were conducted.

The experimental device (Fig. 1) consisted of ejected air receiver 1, jet of active (ejected) gas 2, mixing chamber 3, and outlet diffuser 4. Ejected air was sucked in from the atmosphere into receiver 1 through end-choke washer 5, imitating the hydraulic resistance of the route of the ejected air.

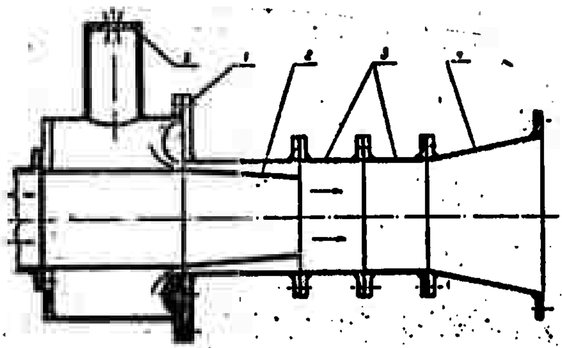


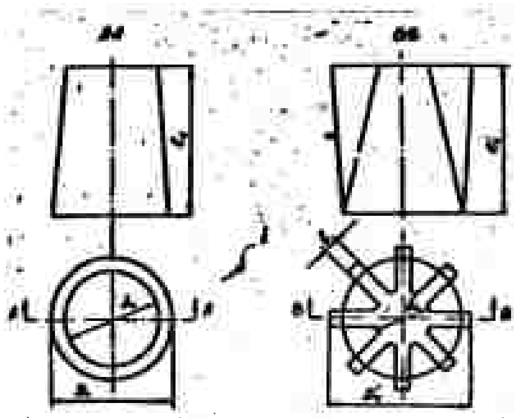
Fig. 1. Diagram of the experimental section.

Condensed air from a compressor entered the active gas jet, to which pressure was applied approximately corresponding to the pressure of the exhaust gases behind the power turbine of a gas-turbine engine.

While conducting the experiments, gas-dynamic measurements necessary for the determination of the flow rate of active G_1 and ejected G_2 air, the coefficient of ejection n, and the resistance of the exhaust system Δp^* were made.

The noise level of the air stream emerging from the ejector was measured by an audio-noise meter of the firm *B&K* of design 2203. The microphone of the audio-noise meter was placed at a distance of 0.5 m from the outlet cross section of the diffusor at an angle of 45° to the axis of the ejector.

Two types of nozzles of active gas (Fig. 2) have been tested. Nozzle *A* had an outlet cross section of annular form. Outlet section *B* consisted of seven rectangular sections with a central round opening.



nozzle А nozzle Б

Fig. 2. Active gas nozzle types:
$l_c = l'_c = 100$ мм, $D_c = 62$ мм, $d_c = 66$ мм, $d'_c = 30$ мм, $D'_c = 96$ мм, $b = 13$ мм.

Both active gas jets were of identical length and had the same ratio of input and output areas $\bar{f}_c = \frac{f_{вх}}{f_{вых}} = 0{,}65.$

The mixing chamber of the ejector consisted of two parts, joined by flanges which allowed us to obtain a mixing chamber with a relative length of $\bar{l}_к = \frac{l_к}{D_к} = 0{,}568$ and 1.136. The relative area of the mixing chamber was $\bar{f}_к = \frac{f_c}{f_к} = 0{,}485;$ the diameter of the mixing chamber was $D_к = 95$ mm.

The circular coniform diffusor, fastened at the outlet from the mixing chamber, had a relative length with respect to the inlet diameter of $\bar{l}_д = \frac{l_д}{D_к} = 1{,}052$ and a relative outlet area of $\bar{f}_д = \frac{f_к}{f'_{вых}} = 0{,}670.$

The central angle of the diffusor opening was $\alpha_д = 12°.$

Adjustment of a device during the tests of all variants of the ejector models was carried out under conditions of maintenance of constant flow rate for each compared variant.

The research was conducted with the temperature of the active air less than the temperature of the exhaust gases of a gas-turbine device. Therefore, the coefficient of ejection obtained on the model, was reduced to the temperature characteristic for the exhaust gases of a gas-turbine engine using the formula [1]

$$n_{пр} = n_{ф}\sqrt{\frac{T_{г}^{*}}{T_{1}^{*}}},$$

in which $n_{ф} = \frac{G_2}{G_1}$ - the actual coefficient of ejection $T_{г}^{*} = 686°K$ - the temperature of the exhaust gases of a gas-turbine engine; T_{1}^{*} - the temperature of the active gas during the tests.

The experimental data was processed with respect to comparative characteristics. As basic variants we adopted ejector variants with an active gas jet of circular outlet cross section (design A). The characteristics of the basic counterparts of the ejector is given in the table.

Table. The parameters of an ejector with an active gas jet of circular outlet cross section.

The flow rate of gas	$G_1 = 0{,}427$ kg/s				$G_1 = 0{,}328$ kg/s				$G_1 = 0{,}244$ kg/s			
Parameter / Code	Δp^* kg/m^2	n %	L db	λ	Δp^* kg/m^2	n %	L db	λ	Δp^* kg/m^2	n %	L db	λ
A2Д	452	4,41	103,3	0,268	273	4,64	97,5	0,210	159	4,63	92,5	0,159
A20	514	3,59	109,3	0,286	321	3,45	103,0	0,228	186	3,33	93,0	0,174
A10	622	1,22	110,0	0,307	383	1,19	101,0	0,249	220	1,34	99,0	0,189

The code in the left column of the Table designates the combination of mixing chamber and diffusor mounting. For example, the code A20 means an ejector with an active gas jet A, with two insets of the mixing chamber, without a diffusor.

The comparative characteristics of the ejector with an active gas jet B is given in Fig. 3. Here $\bar{L} = \frac{L_B}{L_A}$; $\bar{n} = \frac{n_B}{n_A}$; $\overline{\Delta p^*} = \frac{\Delta p^*_B}{\Delta p^*_A}$.

Subscripts A and B refer to the characteristics of an ejector with an active gas jet of design A and B, respectively. A comparison was made with identical values of a given velocity λ in the cross section of the active gas jet.

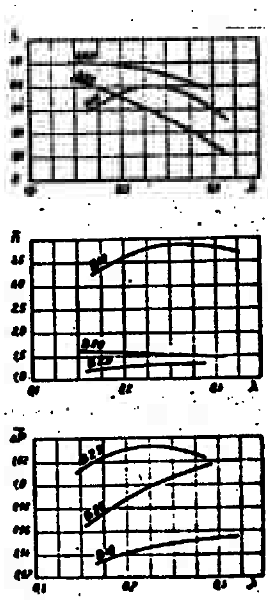


Fig. 3. The effect of the active gas nozzle form on the characteristics of the ejector.

As can be seen from Fig. 3, the application of the active gas section nozzles with the short ejector mixing chambers allows us to considerably decrease the noise level of the stream emerging from the ejector, and to increase the coefficient of air ejection by 1.25-3.75 times with a change in Δp^* from 0.94 to 1.03.

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30 January 1968

ON THE CALCULATION OF THE MAXIMUM DEGREE OF COMPRESSION OF AN EJECTOR

I. I. Kalmykov and I. I. Mosin

In the practice of testing special engines, and also a number of other cases, where ejector devices are used as extractors, it is important to determine the highest degree of rarefaction of the medium in the inlet receiver of the first stage of an ejector or the maximum degree of its compression. As is known, these conditions conform to the work mode of the stage at zero output and maximum pressure drop on the active jet.

The most widespread means for recreating the characteristics of high-pressure ejectors, including the calculation of above-named conditions, are the methods founded upon the one-dimensional theory of a gas flow [1]. Along with their convenience they have a significant shortcoming - poor convergence in the area of small and zero maximum ejection coefficient, especially for an ejector with central feed of the active jet.

As the basis of the existing methods for calculation of the mode with $n = 0$ ($\bar{P} = \bar{P}_{max}$ and $\pi = \pi_{max}$) the following scheme of the gas flow in an initial section is assumed: the jet operates with insufficient expansion during a supercritical pressure drop, in consequence of which the gas, leaving the jet, continues to expand in the initial

section until its external boundary touches the mixing chamber walls; ejection becomes impossible. Such is a one-dimensional analogue of the gas flow under these conditions.

However, in actuality the execution of the basic prerequisite of the theory of this mode - the tangency by the external boundaries of the jet chamber walls - still does not lead to $\overline{n_{pa}=0}$, and the value of n_{np} obtained is more than zero for the calculation value of the drop in pressure at the jets $\overline{p_{pacq}=p_{max}}$.

This bears witness to the fact that an active stream within its boundaries under certain conditions still carries a certain quantity of the passive medium. The analysis of the structure of the corresponding supersonic flow confirms this conclusion.

Figure 1 shows a diagram of the supersonic flow of an axisymmetrical jet, which effuses into motionless space with large degree of noncalculatability $\left(\frac{p_{1a}}{p_{1a}}>2\right)$.

The detailed description of the given diagram and the numbering of the area in Fig. 1 have been taken from work [2].

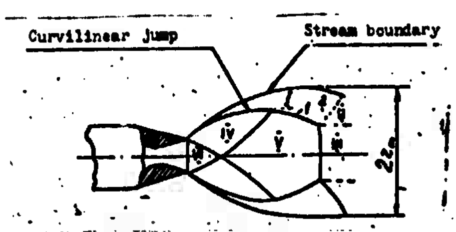

Fig. 1.

Most interesting, from the viewpoint of the issue being discussed, is area I, contained between the boundary of the stream and the

fronts of the branched impact waves 1 and 2. Because of area I the active stream pumps through itself a certain quantity of the ejected medium because of the turbulent mixing along the corresponding stream boundaries.

If we limit this jet by walls, as takes place in an ejector, then as a result of the interplay of the boundary of a supersonic stream with a wall near it reverse flows will arise. Reverse flows[1] will occur at $\bar{n}_{\text{пр}}>0$ and at $\bar{n}_{\text{пр}}=0$. In the latter case, however, the condition of dynamic equilibrium should be fulfilled, namely, the fact that the number of the particles of the ejected media, captured by a stream in area I and returned from it to the line of suction in the form of reverse flows, should be equal.

However, the accurate mathematical description of the cited process with the subsequent solution of equations has so far been difficult, although at the present time definite progress [3] is being made in this direction.

For an evaluation of parameters under conditions $\bar{n}_{\text{пр}}=0$ and $\bar{p}=\bar{p}_{max}$ a well-known distribution was obtained by approximation methods. Below one of these methods is considered.

The following scheme for the gas flow in an initial section under conditions $\bar{n}_{\text{пр}}=0$ and $\bar{p}=\bar{p}_{max}$ is assumed: an active gas effuses from a jet with highly insufficient expansion with a supercritical pressure drop; having continued to expand it again covers the whole transverse cross section of the mixing chamber; but ejection will become impossible only when the boundary turbulent layer of the stream is not forced out, and when the external boundary of the stream does not amount to a certain maximum corner with the wall $\alpha'_{\text{ст}}$ (Fig. 2).

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[1]Reverse flows were observed by us in a clear coniform mixing chamber with the aid of silk threads in the area n = 0-0.05.

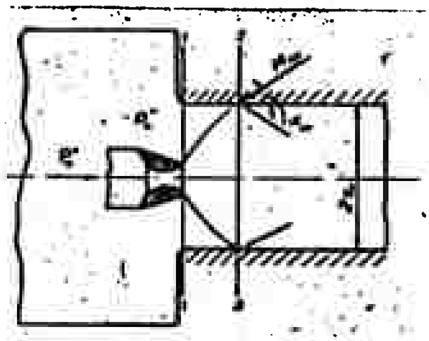

Fig. 2.

To obtain the calculation formulas in the subsequent analysis, conclusion from the work of Prof. I. P. Ginzburg in the section on the theory of gas jets [2] are used.

Despite the sketchiness of the adopted gas flow model in the initial section, considering the complex conditions of the interplay of a supersonic jet with a wall, the quantitative relationships obtained below allow us to obtain data, which agree adequately with the experiment.

The maximum degree of compression for the coniform mixing chamber can be calculated from the known equations of ejection

$$\varepsilon_{max} = \frac{\bar{P}_{max}}{\bar{f}_{sp}\, q(\lambda_3)} \sigma_{nc}\, \sigma_{A}\, \sigma_{3}; \tag{1}$$

$$q(\lambda_3) = \frac{\bar{f}_{21}\, q(\lambda_{2n})}{\bar{f}_{31}}; \tag{2}$$

$$\bar{P}_{max} = \frac{\varepsilon_{sp}}{\pi(\lambda_{1n})} = \frac{1}{\pi(\lambda_{2n})}, \tag{3}$$

where $\bar{P}_{max} = \frac{p^0_{1o}}{p^0_{1n}}$ — the maximum drop in pressure on the active jets;

$\bar{f}_{кр} = \frac{F_1}{F_{кр}}$ — the basic geometric characteristic of the ejector stage;

$\bar{f}_{31} = \frac{F_3}{F_1}$; $\bar{f}_{21} = \frac{F_2}{F_1}$ — the degrees of compression of the mixing chamber and of the initial section, respectively; $F_{кр}$, F_{1a} — is the flow cross-sectional area of an active gas in the critical cross section of a jet and at the outlet from it; F_1 and F_3 - the area of the transverse cross section of the mixing chamber inlet and at the outlet from it; λ_{1a}, λ_{2a} — given velocities for an active gas in cross sections *1-1* and *2-2* (Fig. 2); $\sigma_{п.с}$, $\sigma_{д}$, σ_{a} — the coefficients of pressure recovery of a normal shock subsonic diffusor, and of an active jet, respectively; $n_{кр} = \frac{P_{1a}}{(P_{1a})_{min}}$ — the maximum amount noncalculatability of the stream jet.

However, system (1)-(3) is open-circuited, as the number of the unknown (ε_{max}, $\bar{P}_{max}$, $\lambda_2 \lambda_{2a}$) exceeds the number of equations. Therefore, to system (1)-(3) it is necessary to join a supplementary condition. In accordance with the accepted scheme of the gas flow in the initial section we assume that the external boundary of the jet constitutes a certain maximum angle with the wall $\alpha'_{ст}$. for the estimation of conditions, under which the flow condition with $\alpha_{ст} = \alpha'_{ст}$, will be realized, as a first approximation it is possible to procede from the assumption that under these conditions the passage cross section of the initial section of the mixing chamber and the maximum cross section of the flow nucleus, limited by curvilinear shocks, should be numerically equal. Using the designations in Fig. 3, this condition will be inscribed in the form

$$\bar{R} - |\bar{r}_{к}| = \bar{r}_{г}, \qquad (4)$$

where $\bar{r}_{к} = \frac{r_{к}}{r_{1a}}$ — the relative radius of the mixing chamber.

For a determination of values $\bar{R}$ and $\bar{r}_{к}$ let us make use of the approximation method of calculating the form of the shock wave [2]:

$$
\left.
\begin{aligned}
&\bar{X}_a = \bar{R}\cdot\sin(v_a - \alpha_a);\\
&\bar{r}_a = 1 - \bar{R}\cos(v_a - \alpha_a);\\
&\bar{R} = \frac{\bar{X}_c}{\sin\omega_1 + \sin(v_a - \alpha_a)},
\end{aligned}
\right| \qquad (5)
$$

where $\bar{X}_a = \frac{X_a}{r_{1a}}$; $\bar{r}_a = \frac{r_a}{r_{1a}}$; $\bar{R} = \frac{R}{r_{1a}}$; $\bar{X}_c = \frac{X_c}{r_{1a}}$;

$\bar{X}_c$ – the distance to the location the flywheel disc; v_a – the initial angle of slope of the jetstream; α_a and ω_1 are explained using Fig. 3.

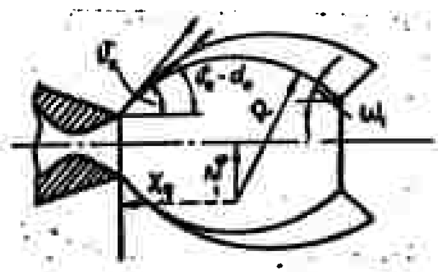

Fig. 3.

The quantities $\bar{X}_c \omega_1 v_a$ and α_a are determined from relationships[1]

$$\bar{X}_c = F(M_c) - F(M_1); \qquad (6)$$

$$\omega_1 = \operatorname{arctg}\frac{1}{\sin v}\left[\frac{V}{V_1} - \cos v\right]; \qquad (7)$$

---

[1]When $n > 2$ the value of $\bar{X}_c$ can be found from the simpler empirical Love's formula [4]:

$$\bar{X}_c = 0{,}8\left\{3{,}1\,M_{1a}^{1/3}\left[(nM_{1a}^2 - 1)^{1/2} - (M_{1a}^2 - 1)^{1/2} - 2\sqrt{M_{1a}^2 - 1}\right]\right\}\left(\frac{n}{2}\right)^{t},$$

where $t = 0{,}451 - 0{,}016M_{1a}$.

$$v_n = v_s + \omega(M_n) - \omega(M_{1n}); \tag{8}$$

$$\alpha_n = \arcsin \frac{1}{M_n}. \tag{9}$$

where

$$F(M) = \frac{3-k}{k-1}\sqrt{M^2-1} - \frac{2}{k-1}\sqrt{\frac{k+1}{k-1}} \operatorname{arctg} \sqrt{\frac{k-1}{k+1}(M^2-1)};$$

$$\omega(M) = \sqrt{\frac{k+1}{k-1}} \operatorname{arctg} \sqrt{\frac{k-1}{k+1}(M-1)} - \operatorname{arctg}\sqrt{M^2-1};$$

$$v = \arccos \frac{\left[\left(\frac{1-1/\varepsilon}{1-1/\bar{\varepsilon}}\right)\frac{V_1}{V}\cdot\frac{p_1}{p} V_1 - V\right]}{\left[\left(\frac{1-1/\varepsilon}{1/\bar{\varepsilon}-1}\right)\frac{V_1}{V}\cdot\frac{p_1}{p} V_2 - V_1\right]}.$$

Here $\varepsilon = \frac{p_1}{p}$; $\bar{\varepsilon} = \frac{p_1}{p_2}$; p_1, p_2 — pressure in the areas I and II; p - the pressure before the normal shock wave; $M_1 M_c M_n$ — Mach number values up to a normal shock when $x=0$ and $\bar{r}=1$, and when $x=x_c$ at the boundary of the jetstream respectively, are determined from equations

$$\frac{\left(1+\frac{k-1}{2}M_c^2\right)^{\frac{k}{k-1}}}{\frac{2k}{k+1}M_c^2 - \frac{k-1}{k+1}} = \varepsilon\left(1+\frac{k-1}{2}M_{1a}^2\right)^{\frac{k}{k-1}}; \tag{10}$$

$$M_1^2 = 1 + \frac{k+1}{k-1}\operatorname{tg}^2\left\{\sqrt{\frac{k-1}{k+1}}\left[\frac{\pi}{2} + \omega(M_{1a}) - v_a\right]\right\} \tag{11}$$

and

$$\left(1+\frac{k-1}{2}M_n^2\right)^{\frac{k}{k-1}} = \varepsilon\left(1+\frac{k+1}{2}M_{1a}^2\right)^{\frac{k}{k-1}}. \tag{12}$$

The system of equations (5)-(12) can be solved by the method

of sequential approximation. The velocity value at the boundary of the jetstream $M_{ж}$ which satisfies condition (4), is the desired one. Assuming $M_{ж} = M_{3ж}$, according to scaling formulas we determine the value of $\lambda_{3ж}$. For the quantity $\lambda_{3ж}$ with the aid of equations (1)-(3) $\bar{p}_{max}$, will be located, and then ε_{max}.

To check the reliability of the calculation procedure for the maximum degree of ejector compression an experiment was carried out. The experimental device was a two-stage ejector with central positioning of the active nozzles. Maximum pressure in an active main line was as high as 7 [atm(abs.)], and minimum pressure in the main line of the passive medium was as low as 0.01 [atm(abs.)]. The flow rates of the active and passive media were determined using standard procedures. The pressure in the active main line was measured with the aid of a specimen pressure indicator with a scale value of 0.02 kg/cm^2; the pressure in the main line of the passive medium was measured at more than 20 mm Hg with a mercury piezometer, but at less 20 mm Hg with an oil piezometer. In the latter case, to one of the elbows of the piezometer a mechanical vacuum pressure device RVN-20 was connected. The temperature in that and another main line was determined with the aid of temperature gauges with a scale value of 0.2°.

A change in the basic geometric characteristics of the investigated stage of the ejector with $f_{кр} =$ 5.9 and 17 was attained by replacement of the active nozzles. The latter were made with three different values of critical diameter ($d_{кр} =$ 10.2, 13.2, and 17 mm).

The mixing chamber at the inlet had an initial cylindrical section with a length of 2 calibers ($D_1 = 42$ mm), a transient coniform section with an angle of conicity of $2\gamma = 15°$ and degree of compression $\bar{f}_{21} = \frac{F_2}{F_1} = 0,6$.

From the total measured pressures of an active and passive gas at the inlet to the ejector and of the combined flow at its outlet in the maximum mode the drop in pressures on the active nozzles and the degree of compression were determined.

A comparative evaluation of the convergence of the calculation procedure of conditions when $a_{cp}=0$ was made with experience from the magnitude of the maximum drop in pressure $\bar{P}_{max}$, but not from the magnitude τ_{max}. As a basis for this served the following circumstance: entering into formula (1) for a determination of τ_{max} are a number of experimental coefficients $e_A e_B$, during the selection of which a certain arbitrariness is admitted; but formula (3) lacks this deficiency for a determination of $\bar{P}_{max}$; the quantity $\bar{P}_{max}$ is a function of only one and at that the characteristic variable $\bar{\lambda}_{m}$, which is the desired quantity in the solution of system (4)-(12). Between values τ_{max} and $\bar{P}_{max}$ there is a single-valued connection in the form of a relationship (1). Therefore, from the quantity $\bar{P}_{max}$ it is possible to judge with sufficient completeness the reliability of the method, and also the correctness of the simplifying prerequisites assumed in it.

Figure 4 gives the results of the experiment, and also the results of the calculation $\bar{P}_{max}$ from formulas (3)-(12).

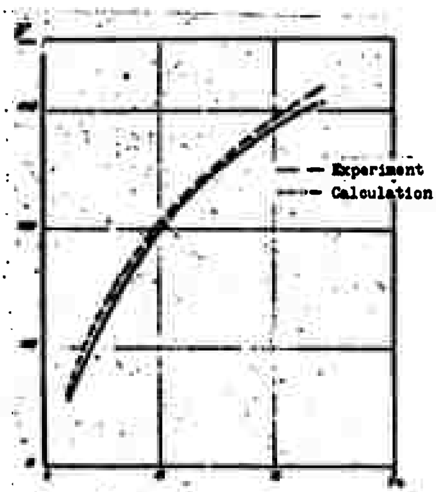

Fig. 4.

Figure 4 indicates that the calculation data adequately agree with the appropriate experimental data.

The resulting divergence in the values of $\bar{p}_{max}$ does not exceed 3-10% in the entire investigated range of values of the basic geometric characteristics of the stage $\bar{f}_{cp} = 5{,}9 - 17{,}0$.

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1 February 1968

THE RATIONAL METHOD OF SELECTING PARAMETERS AND CALCULATING SINGLE-STAGE GAS TURBINES

V. A. Strunkin

The methods of calculating gas turbines [GTD] (ГТД) published in the literature have a number of substantial deficiencies. Randomly (within a recommended, but rather wide range) a large quantity of the parameters of a stage is selected; as a result of which, as a rule, there is obtained the nonconformity of some definable parameters of the stage to the requirements shown for them (for example, a large negative reactivity or a great difference in the angles $\beta_2 - \beta_1$ in a root cross section, or an unacceptable ratio of the height of nozzle and that of the working blades, etc., is inadmissible). This requires the execution of a whole series of calculation variants and a large calculation effort. The mission of producing a turbine of the smallest possible diametric size, which is one of the main problems for GTD, is not considered.

The procedure being proposed is to a considerable degree free of the deficiencies shown.

1. In the design of gas turbines we know the total pressure p^*_0 the bar and the temperature of the gas T^*_0 °K at the outlet; the gas flow rate G, kg/s; internal power N_i, kW, and the rpm n, equal to the rotations of the compressor.

2. We determine the values of the following parameters:

$$A=\frac{L_i}{\frac{k}{k-1}RT_0^*}=0{,}87\frac{N_i}{GT_0^*};$$

$$B=\frac{Gn^2\sqrt{T_0^*}}{10^{10}\cdot p_0^*}.$$

Here L_i — the internal work; $k = 1.33$; $R = 288.3$ J/kg·deg.

3. Let us select a given velocity at the turbine outlet λ_{a_2}, taking into account that with its increase the elongation stresses in the blades are decreased, but, on the other hand, the operation of the output equipment of the engine is impaired. Usually $\lambda_{a_2} = 0.5$-0.7 in a turbojet engine and up to 0.85 in a turboprop engine.

Using the equation of the flow rate, the introduced parameters A and B, and also the formula for stresses [1]

$$\sigma=0{,}1\frac{u^2}{\varphi}\ \frac{\text{daN}}{\text{cm}^2} \qquad (1)$$

it is possible to obtain the following expression:

$$\sigma=221\frac{1}{\sin\alpha_2}\cdot\frac{1}{q(\lambda_{a_2})}\cdot B\frac{\sqrt{1-A}}{(1-A/\eta_i^*)^{\frac{k}{k-1}}}, \qquad (2)$$

where α_2 — the output angle of the flow from a stage; η_i^* — inner efficiency of the stage.

Subsequently, the value σ/c_2^2 for which we can derive the expression

$$\frac{\sigma}{c_2^2}=\frac{\sigma}{18{,}15^2 T_0^*(1-A)\lambda_{c_2}^2}, \tag{3}$$

since

$$c_2=18{,}15\sqrt{T_0^*(1-A)}\,\lambda_{c_2}. \tag{4}$$

For equations (2)-(4) a nomograph (Fig. 1) has been constructed, which allows us to rapidly find from the selected quantity λ_{c_2} the velocity c_2, stresses σ, and the ratio σ/c_2^2. In the creation of the nomograph it is assumed that $\eta_t^*=0{,}9$; $\alpha_2=85°$. Taking into account that when $\alpha_2=70-110°$ $\sin\alpha_2\approx 1$, this nomograph can be used with any α_2 within the limits shown. It may appear that even for the greatest value of λ_{c_2} (within the boundaries shown) the stresses σ are inadmissibly high. In this instance it is necessary to decrease the rotations, having selected in the calculation of the compressor the smaller value of the peripheral velocity at the circumference u_k.

Final judgment about the acceptability of stresses can be made after the determination of the blade temperature using formula (11).

4. Let us make the choice of a peripheral velocity u at an average diameter in such a way that under the root cross section of a stage the condition

$$\vartheta_{1k}=\vartheta_{2k} \tag{5}$$

would hold.

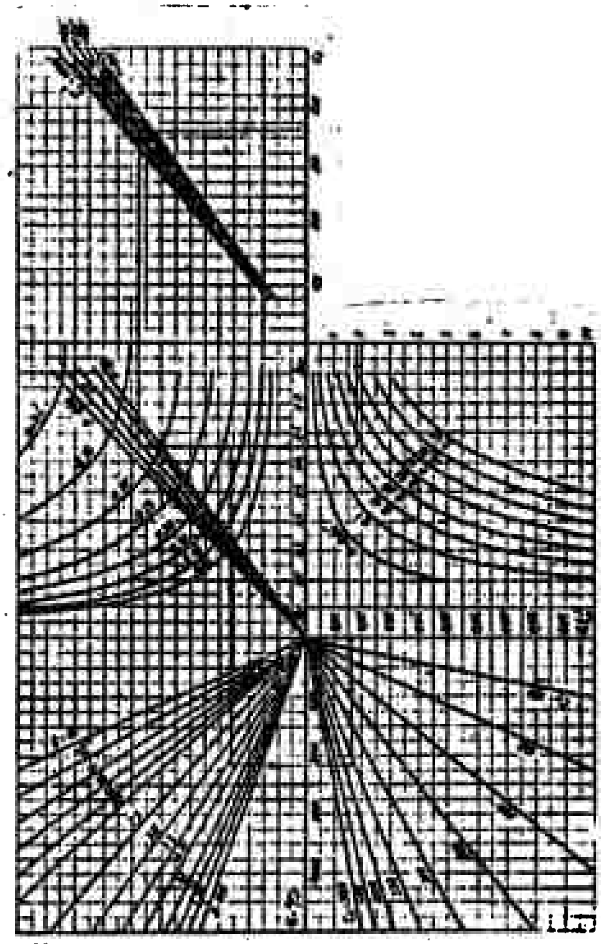

Fig. 1. Nomograph No. 1.

It is obvious that the circular velocity and the over-all size of the turbine (at fixed rotations) in this instance will be minimum. Below on 7 the possibility of a further decrease of u and the over-all size of the turbine is examined.

If we use condition (5), the dependence between parameters in the stage, twisted according to the law $rc_u = \text{const}$ and $c_a = \text{const}$, formula (1), the formula for circular operation

$$L_u = u(c_{1u} + c_{2u}) \qquad (6)$$

and the assumption that

$$\left(1 - \frac{1}{\varphi}\right)^2 \approx 1 - \frac{2}{\varphi},$$

it is possible to obtain the following equation for the determination of the sought for value of peripheral velocity:

$$\left(\frac{u}{c_2}\right)^2 + \left(\frac{u}{c_2}\right)\cos\alpha_2 = \frac{mM}{1+a} + 20\frac{\varepsilon}{c_2^2}. \qquad (7)$$

These designations are introduced:

$$M = \frac{L_1}{c_2^2} = \frac{A}{0{,}283\,(1-A)\lambda_{c_2}^2}; \qquad (8)$$

$$m = \frac{L_2}{L_1}; \qquad (9)$$

$a = \frac{c_{1a}}{c_{2a}}$ — the relationship of axial component velocities c_1 and c_2.

The quantity N on selected λ_{c_1} and assigned a can be determined from nomograph 1 (Fig. 1).

Equation (7) is graphically depicted by nomograph 2, which allows us to rapidly find the value of $\frac{u}{c_1}$ and the peripheral velocity u. In the construction of the nomograph it was assumed that m = 1.02.

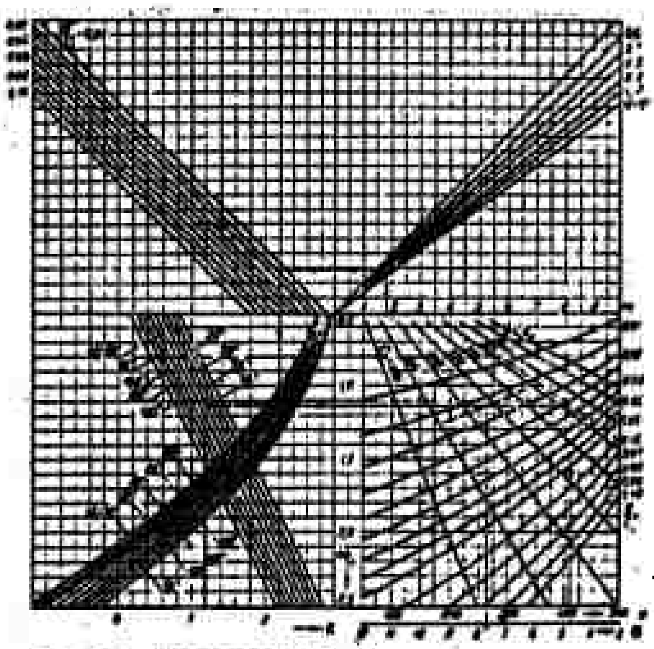

Fig. 2. Nomograph No. 2.

Analyzing the nomograph, we see that a reduction in the angle α_2 noticeably affects the quantity u. The lower is α_2, the lower u is, and therefore, the over-all size of the turbine. The effect of α_2 is especially strong at high velocities of c_2, i.e., with large λ_{c_2}. However, when selecting α_2 one ought to consider the optimum operating conditions of a GTD jet nozzle. Therefore, it is usually assumed that $\alpha_2 = 82-85$.

Nomograph 2 allows us to find also the quantity $\psi=\frac{a}{\xi_0}$ (lower righthand quandrant).

For calculations with the aid of nomograph 2 or from equation (7) it is necessary to select the quantity a. From the nomograph it is evident that the greater the value a, the smaller u and the over-all size of the turbine. However, with an increase in a, in the first place, the reactivity in the root cross section is decreased ρ_k. With condition (5) for ρ_k the following formula can be derived:

$$\rho_k=\frac{c_{2u}^2\left(\frac{1}{\psi^2}-a^2\right)\left(\frac{1}{\operatorname{tg}^2\beta_{2k}}+1\right)}{c_{ad}^2}. \qquad (10)$$

From this formula it is evident that when $a>\frac{1}{\psi}$ the reactivity ρ_k becomes negative.

In the second place, one ought to take into account the fact that with an increase a the height of nozzle lattice l_1, is decreased, in comparison with the working l_2. This can lead to an inadmissibly large angle of opening of the flow section. Calculations indicate that the magnitude a should be selected within the limits of 0.7-0.9. If, besides the rotations, for any reasons the diameter of the turbine is also given, then the peripheral velocity becomes known and the necessity in stage No. 4 for calculation ceases.

5. The value $\frac{u}{c_0}$ obtained in the previous stage of calculation allows us to determine the temperature of the metal of a working blade [1]

$$T_A=0{,}95\,T_w^*\approx0{,}95\left(T_2^*+\frac{w^2}{2\frac{k}{k-1}R}\right)=$$

$$=0{,}95T_0^*\left[1-A+\frac{A}{2M}\left(\frac{u}{c_0}\right)^2\right]. \qquad (11)$$

Known values σ, T_2 and the given life span of the blades allow us to select the material for the working blades and to calculate the safety factor, which should take account of the presence of bending stresses, which are not determined in the given method.

6. For a determination of the ratio $\frac{l_1}{l_2}$ it is first necessary to find the value of reactivity ρ at an average radius.

Let us transform the expression for reactivity

$$\rho = \frac{\frac{w_2^2}{\psi^2} - w_1^2}{c_{ad}^2}. \qquad (12)$$

The rate of the adiabatic discharge, which corresponds to work L_0.

$$c_{ad}^2 = 2L_0 = 2L_0^* + c_0^2 \frac{T_{2s}^*}{T_2^*}. \qquad (13)$$

Here

$$L_0^* = \frac{L_1}{\eta_t^*}; \qquad (14)$$

$$T_{2s}^* = T_0^* - \frac{L_0^*}{\frac{k}{k-1}R} = T_0^*\left(1 - \frac{A}{\eta_t^*}\right);$$

$$T_2^* = T_0^* - \frac{L_1}{\frac{k}{k-1}R} = T_0^*(1-A). \qquad (15)$$

Since

$$w_2^2 = c_{2a}^2 + (u + c_{2u})^2;$$

$$w_1^2 = a^2 c_{2a}^2 + \left(\frac{mL_1}{u} - c_{2u} - u\right)^2.$$

then, using adduced formulas, we will find

$$\rho = \frac{\left(\frac{1}{\psi^2} - a^2\right)\sin^2\alpha_2 + \frac{k^2}{\psi^2} - \left(\frac{mM}{u/c_2} - k\right)^2}{2M/\eta_i^* - \frac{1 - A/\eta_i^*}{1 - A}}, \tag{16}$$

where

$$k = \frac{u}{c_2} + \cos\alpha_2.$$

From formula (16) it is possible to calculate ρ for a selected value a. The calculation is simplified by nomograph 3 (Fig. 3), built from formula (16) with $\psi = 0{,}97$; $m = 1{,}02$; $\alpha_2 = 85^\circ$ and $\eta_i^* = 0{,}9$. The quantity k can be determined from nomograph 2 (lower left quadrant).

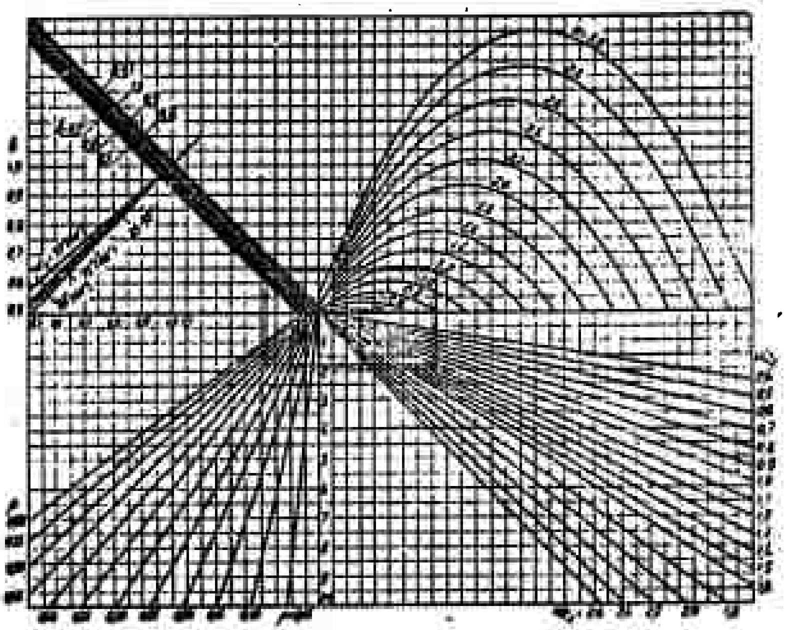

Fig. 3.

If angle α_2 noticeably diverges from 85°, then in using nomograph 3 it is first necessary to find from the graph, given in the upper left quadrant, the value of the parameter b, which depends upon the coefficient a and angle α_2. Quantity b is determined from the condition

$$\left(\frac{1}{\varphi^2}-b^2\right)\sin^2\alpha_2=\left(\frac{1}{\varphi^2}-a^2\right)\sin^2 85^\circ.$$

At $\alpha_2=85-95^\circ$ this parameter practically coincides with quantity a.

In the formulation of nomograph 3 it was taken into account that quantity ρ practically does not depend upon the value of parameter A within the limits of $A=0-0.4$. Therefore, the second term in the denominator of (16) is taken as equal to 0.97.

7. The ratio of the heights of the nozzle and of the working lattices we can find from the equation of the flow rate, assuming an average diameter and flow rate through them to be identical

$$\frac{l_1}{l_2}\cdot a=\left[\frac{1-r}{1-(1-\rho)r}\right]^{\frac{k}{k-1}}\cdot\frac{1-\varphi^2(1-\rho)r}{t}. \qquad (17)$$

Here we introduced the designation

$$r=\frac{A}{\varphi^2}+\frac{A}{2M}\cdot\frac{1-\frac{A}{\varphi^2}}{1-A};$$

$$t=1-A-\frac{A}{2M}.$$

From equation (17) we constructed nomograph 4 (Fig. 4), considerably simplifying the calculations. It is assumed that $\varphi = 0{,}96$.

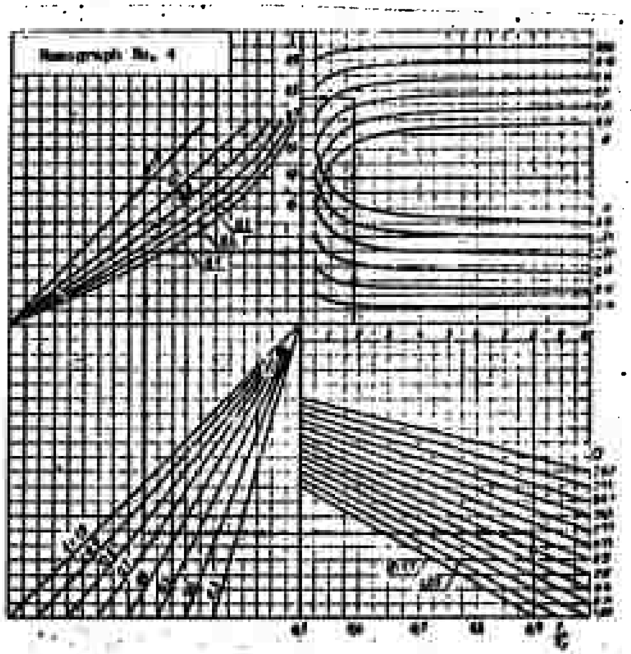

Fig. 4.

Taking into account that the angle of opening of the flow section, it is possible to a certain degree to regulate by structural measures the minimum height ratio $\frac{l_1}{l_2} = 0{,}9 - 0{,}8$. If this obtained relationship was inadmissibly small, then one ought to repeat the calculation using formulas (7), (16) and (17) or nomographs 1-4, after selecting a smaller value for *a*.

8. To decrease the over-all size of a turbine, it is possible to digress from condition (5), admitting angles $\beta_{1\text{л}}$ somewhat smaller (up to 10°) than $\beta_{2\text{л}}$.

$$\beta_{2k} - \beta_{1k} = \Delta\beta_k, \qquad (18)$$

where $\Delta\beta_k$ is small in comparison with β_{2k}.

Let us keep as invariable the values c_{1u}, c_{2u} (i.e., α), c_{2k}, β_{2k}.

Let us take into account the fact that the circular operation should be identical with condition (5) and with condition (18), whence

$$u_k(c_{1uk} + c_{2uk}) = u'_k(c'_{1uk} + c'_{2uk}). \qquad (19)$$

The dashed lines in equation (19) and in Fig. 5 designate the parameters of the velocity triangles with condition (18).

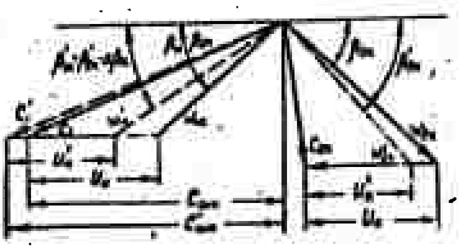

Fig. 5. Triangles of velocities in a root cross section.

We assume that

$$u' \approx u(1 - \varepsilon\Delta\beta_k). \qquad (20)$$

Using the apparent relationships between the elements of the triangles of velocities, dependence (20) and assuming angle $\Delta\beta_k$ has a small value, from equation (19) it is possible to obtain the following formula for the coefficient ε:

$$\varepsilon = \frac{a}{1+a}\cdot\frac{\sin\alpha_2+\frac{1}{\sin\alpha_2}\left(\frac{u}{c_2}\cdot\frac{\theta-1}{\theta}+\frac{\theta}{\theta-1}\cos\alpha_2\right)^2}{2\frac{u}{c_2}\cdot\frac{\theta-1}{\theta}+\frac{\theta}{\theta-1}\cos\alpha_2}. \qquad (21)$$

This formula includes the values obtained in the calculation with condition (5).

Thus, selecting $\Delta\beta_k < 10^{\circ}$, from formula (20) taking into account (21), it is possible to find the new value of the peripheral velocity u', which is smaller than the earlier found u.

New values ρ and $\frac{l_1}{l_2}$ can be found from equations (16) and (17) or nomographs 3 and 4, preliminarily having determined the new value of the ratio $\left(\frac{u}{c_2}\right)'$ (c_2 = idem):

$$\left(\frac{u}{c_2}\right)' = \left(\frac{u}{c_2}\right)(1-\varepsilon\Delta\beta_k).$$

One ought to keep in mind that the reactivity in this instance is decreased, while the ratio $\frac{l_1}{l_2}$ increases.

For the convenience of calculations we constructed nomograph 5, from which it is easily possible to find the ratio of the peripheral velocities $\frac{u'}{u}$ or the quantity $\left(\frac{u}{c_2}\right)' / \left(\frac{u}{c_2}\right)$. Here auxiliary quantity $C = \frac{\theta}{\theta-1}\cos\alpha_2$.

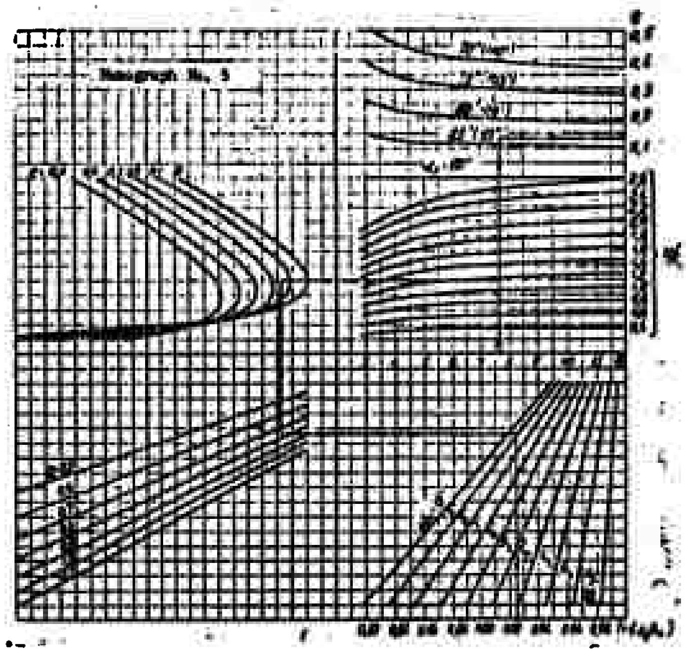

Fig. 6.

9. Further, from the found quantity u (or u', if we assume condition (18)) we will determine values c_{1u}, c_1, α_1 and β_2, which together with the known ρ and θ allows us to find the loss due to blow-by Z_{yr} and the quantity of gas passing through working blades $G_a = G - G_{yr}$, according to methods given in work [1]. Knowing Z_{yr} and L_l, let us find the more precise value of peripheral operation L_u, because from formula (9) it was found to be approximately

$$L'_u = L_l + Z_{yr}. \qquad (22)$$

If the obtained value L'_u is less than that assumed earlier in (9), one ought to decrease the peripheral velocity with respect to $\frac{L'_u}{L_u}$. In this case the average and, therefore, the overall diameter

of the turbine is decreased. In the other case, when $L_u' > L_u$, one ought to modify the value c_{1u}, having determined it according to formula

$$c_{1u}' = \frac{L_u'}{u} - c_{2u}.$$

Velocities u and c_{2u} in this instance we leave without changes.

10. After the introduced amendments, which ensure the obtaining of assigned operation L_i, it is possible to make the final calculation of all the parameters of the stage, having selected the design of the following section.

In this case in series we determine the values $c_1, \alpha_1, w_1, \beta_1, w_2, \beta_2, p_1, T_1, p_2, T_2, l_1, l_2, \rho$ and the efficiency of the stage

$$\eta_i^* = \frac{2L_i}{c_{ad}^2 - c_2^2},$$

where

$$c_{ad}^2 = 2L_0 = 2\frac{k}{k-1}RT_0^*\left[1 - \left(\frac{p_2}{p_0^*}\right)^{\frac{k-1}{k}}\right].$$

In the majority of published methods for calculation of gas turbines at a non-coincidence of the obtained (as a result of enormous variant calculations!) and assumed values of the efficiency the conducting of a final calculation is required.

In this case the obtained values of the efficiency is final,

even if it did not coincide with the initially assumed quantity $\eta_t^* = 0.9$. This is explained by the fact that the obtained parameters of the stage ensure the assigned operation L_t^*.

As an illustration, on the nomographs the calculation of the turbine with the same parameters as in work [1] is shown.

11. The method given here and, especially, the nomographs can also be used in the calculation of multistage turbines. This will be indicated in the following work of the author [2].

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A GENERALIZATION OF THE RESULTS OF MEASUREMENTS OF THE FINENESS OF FUEL ATOMIZATION OF MECHANICAL AND AIR-MECHANICAL INJECTORS OF THE PRESSURE-JET TYPE

I. N. Dyatlov

In this work we have given the results of the mathematical processing of experimental analyses to determine the fineness of atomizations mechanical and air-mechanical pressure-jet injectors axifugal.

At the same time an analysis of the existing formulas has been conducted to determine the fineness of atomization and the possibility of their application to our atomization conditions has been investigated.

Generalizations were made for a axifugal FR-3 injector with mechanical fuel atomization and for a fuel-air injector (TVF-1).

The fuel-air injector [1, 2], without supplying atomizing air to it, is an ordinary centrifugal injector with mechanical atomization.

The fineness of the atomization TVF-1 was determined both with and without feeding atomizing air to it. Checks were conducted for two types of fuel: for aviation kerosene T-1 (GOST 4138-49) and diesel-engine tractor fuel (GOST 385-42).

1. The Determination of Fineness with Mechanical Fuel Atomization

Until now no one succeeded in finding a theoretical solution to the issue of fineness of fuel atomization by centrifugal injectors. This is explained by the complexity of occurring processes and by the great diversity of the construction features of atomizers.

Therefore, in works dedicated to research on the quality of atomization, either empirical formulas, which are adequate only for the studied types of discharging jets, or formulas obtained by the method of simulitude and dimensionality are used. The latter somewhat extend the area of their application.

As the determining dimension in a number of works the diameter of the output nozzle jet is used.

In general form this dependence can be represented in the following manner [3]:

$$\frac{d}{d_c} = f\left(A, \frac{D_\kappa}{d_c}, \frac{h}{d_c}, \mathrm{Re}, \frac{\mu_T^2}{\rho_T \sigma_T d_c}, \frac{\rho_T}{\rho_\Gamma}\right), \qquad (1)$$

where d – the average diameter of a drop; d_c – the diameter of the nozzle jet; D_κ – the diameter of the swirl chamber; h – the height of the swirl chamber; Re – the Reynolds number; μ_T – the dynamic viscosity of the fuel; ρ_T and ρ_Γ – the density of the fuel and of the gas; σ_T – the surface tension of the fuel; A – the geometric characteristics of the injector.

Experiments conducted by a series of authors indicate that parameters $\frac{D_\kappa}{d_c}$, $\frac{h}{d_c}$, and $\frac{\rho_T}{\rho_\Gamma}$ are weakly affected by a change $\frac{d}{d_c}$, therefore equation (1) can be written in the form of the following exponential complex:

$$\frac{d}{d_c} = CA^{a}\,\mathrm{Re}\left(\frac{\mu_T^2}{\rho_T \sigma_T d_c}\right)^{b}. \qquad (2)$$

Investigating the fineness of atomization of centrifugal injectors, A. G. Blox and E. S. Kickina [4] offered the generalized formula

$$\frac{d}{d_c}=47{,}8\,A^{-0{,}6}\left(\frac{\mu_T^2}{\rho_T\,\sigma_T\,d_c}\right)^{-0{,}7}\left(\frac{v d_c}{\nu_T}\right)^{-0{,}7}, \qquad (3)$$

where v – the rate of fuel discharge; ν_T – the kinematic viscosity of the fuel.

Processing of the experimental data in work [5] allowed us to establish a dependence for the determination of the average diameter of drops

$$\frac{d}{d_c}=3{,}38\,A^{-0{,}1}\left(\frac{\mu_T^2}{\rho_T\,\sigma_T\,d_c}\right)^{-0{,}36}\mathrm{Re}^{-0{,}14}. \qquad (4)$$

Generalizing the experimental data of a number of authors, in research work [6] there is given the equation, which has the form

$$\frac{d}{d_c}=11{,}5\,\mu^{0{,}91}\left(\frac{\mu_T^2}{\rho_T\,\sigma_T\,d_c}\right)^{-0{,}24}\mathrm{Re}^{-0{,}7}, \qquad (5)$$

where $\mu=f(A)$.

Tet and Marshall [7] give the dependence of the average diameter upon the ratio to tangential and axial rates of discharges of the liquid from the nozzle of the indicator

$$\frac{d}{d_c}=2{,}9\left(\frac{v_T}{v_э}\right)^{-0{,}5}\left(\frac{\mu_T^2}{\rho_T\,\sigma_T\,d_c}\right)^{-0{,}24}\mathrm{Re}^{-0{,}7}, \qquad (6)$$

where $v_э$ – the equivalent velocity at the outlet jet; v_T – the tangential, component of velocity at the outlet from the eddy channels.

Investigating the dissociation of a layer of liquid into drops, I. I. Novikov [8] gives an approximation solution for the determination of an average drop diameter in the atomization of a liquid by a centrifugal injector

$$d = \frac{3}{\sqrt[3]{4}} \sqrt[3]{\frac{\sigma_T r^2_{вх}}{P_T R^2}}, \tag{7}$$

where P_T – the pressure of the fuel; R – the radius of the chamber of a swirler; $r_{вх}$ – the radius of input openings into the swirl chamber.

Generalizing the results of the experimental analyses on the fineness of atomization of paraffin by a centrifugal injector, N. N. Strulevich [9] obtained a formula

$$d = \frac{r_c\left(1-\sqrt{1-\mu\cos\frac{\alpha}{2}}\right)}{0{,}11\mathrm{Re}^{0{,}34}\cdot\cos\frac{\alpha}{2}}, \tag{8}$$

where μ – the coefficient flow rate; α – the atomization angle; r_c – the radius of the nozzle jet.

In work [10] a generalization of the experimental data of various authors is given on the analysis of the fineness of atomization by a centrifugal injector. A generalization was made with the aid of dimensionless criteria and an equation was derived;

$$\lg\frac{d}{\delta} = k\left(\frac{\delta\rho_ж\sigma}{\mu^2_ж}\right)^{-0{,}133} - 0{,}35\lg\left(\frac{U^2\rho_ж\delta}{\sigma}\right). \tag{9}$$

Here δ – the thickness of a liquid layer; k – a coefficient, equal to 4.47 for water and aqueous glycerine solutions and 2.9 for kerosene and melted paraffin.

Longvell [6] offered the following formula:

$$\frac{d}{d_c} = \frac{0{,}135\,\sigma^{0{,}7}}{P_T\sin\frac{\alpha}{2}}. \tag{10}$$

In work [12] results of the generalization of experimental data on the atomization of water, kerosene, and benzine in a criterial form have been given

$$\frac{d}{\delta}=\left(135+3{,}67\cdot 10^{-3}\cdot\frac{\sigma_{т}\,\delta\,\rho_{т}}{\mu_{т}^{2}}\right)\left(\frac{v_{т}\,\delta\,\rho_{т}}{\mu_{т}}\right)^{-0{,}9}. \qquad (11)$$

In mechanical fuel atomization with FR-3 and TVF-1 injectors results of our experimental analyses were generalized by the following dependence:

$$\frac{d}{d_{c}}=2{,}35\left(\frac{\mu_{т}^{2}}{\rho_{т}\,\sigma_{т}\,d_{c}}\right)^{-0{,}18}\cdot\left(\frac{v_{т}\,d_{c}}{\nu_{т}}\right)^{-0{,}475}. \qquad (12)$$

For a comparison Figs. 1 and 2 give the dependence $\frac{d}{d_c}=f(P_т)$, obtained by an experimental method and calculated using various formulas for FR-3 and TVF-1 injectors (without supplying atomizing air to the latter).

From these figures it follows that the results of our experiments are adequately described by equation (12).

Comparing the results of our experiments with the calculation data of other authors, we see that the closest results for the investigated discharge injectors occur in the calculation $\frac{d}{d_c}=f(P_т)$ from formulas (3) and (9), while the formulas of other authors show significant divergence.

Thus, despite the great diversity of calculation formulas for the determination of the average diameter, as a rule, they have a particular character. Therefore, they cannot be considered generalized for all types of centrifugal injectors with mechanical fuel atomization.

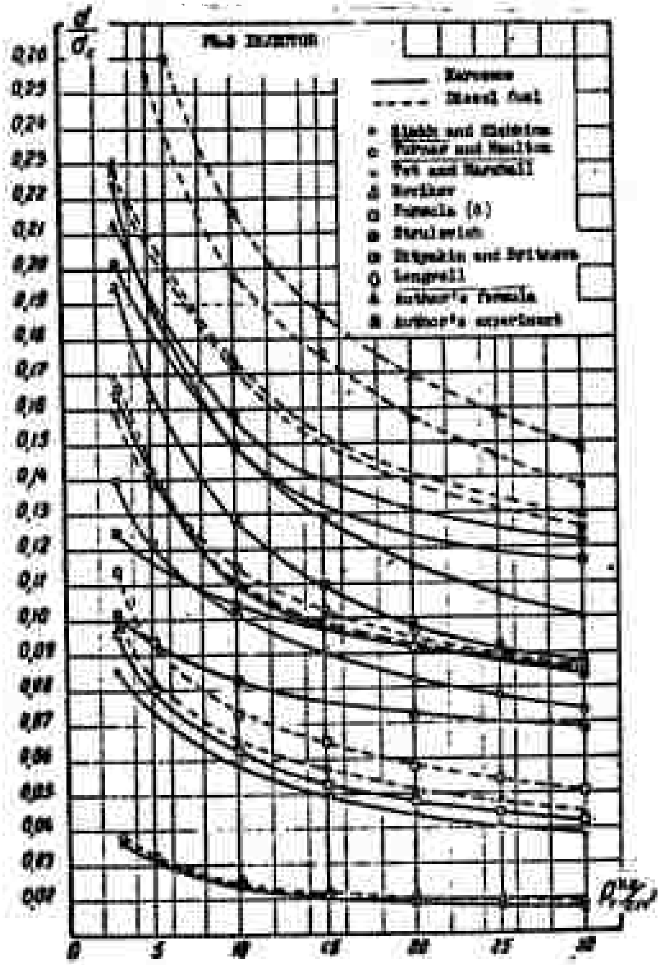
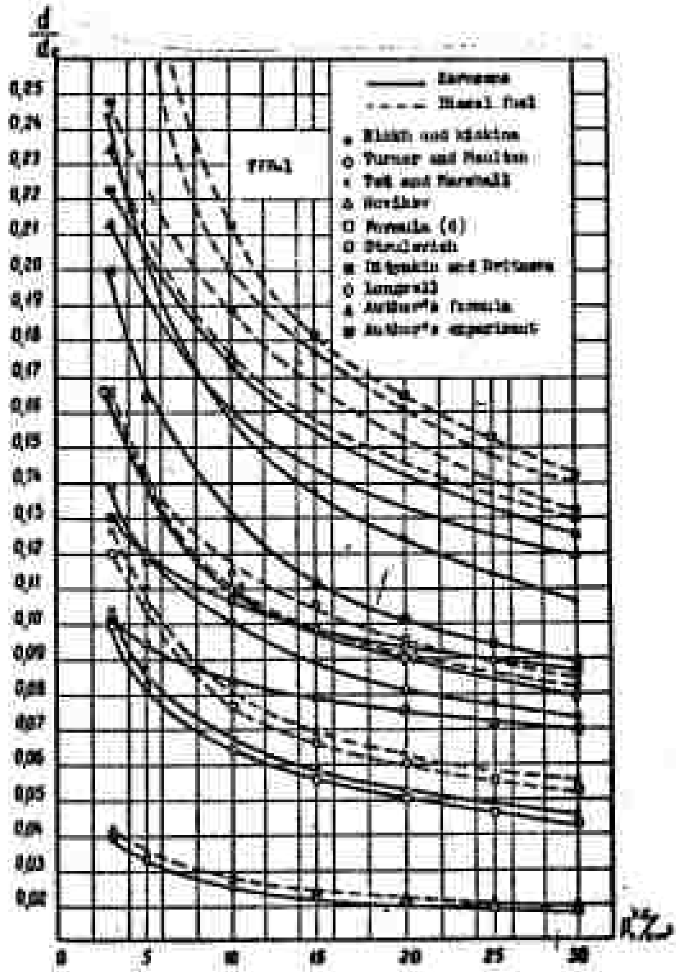


Fig. 1.



d/d_c
0,25
0,24
0,23
0,22
0,21
0,20
0,19
0,18
0,17
0,16
0,15
0,14
0,13
0,12
0,11
0,10
0,09
0,08
0,07
0,06
0,05
0,04
0,03
0,02
Diesel fuel
Turner and Moulton
Author's formula
Author's experiment
0
5
10
15
20
25
30

Fig. 2.

They are applicable (just as is equation (12)) only for the investigated type of discharge injectors in the studied range of change in the parameters of the process.

2. The Determination of the Degree of the Fineness with Air and Air-Mechanical Fuel Atomization

The dissociation of a jet of liquid during air and air-mechanical fuel atomization has been studied less than dissociation during mechanical atomization.

In a few works dedicated to this issue, experimental material on the determination of the fineness of atomization is generalized either by a criterial dependence, or using empirical formulas.

For example, during the analysis of the atomization of water by air nozzles of pressure-jet type in the work [13] the method of similitude was used. The results of all the series of experiments were generalized by the approximation dependence

$$d = C\frac{2\sigma_{ж}}{\rho_{г}} \cdot \left(\frac{d_{c}}{\nu_{г}}\right)^{1,4} v_{от}^{-0,8}, \tag{13}$$

where $v_{от}$ – the relative velocity of air and of liquid at the site of their encounter; C – an experimental constant which depends upon the design of the atomizer.

The method of similitude was used even in the analysis of the fineness of atomization by low-pressure air injectors [14, 15].

The results of experimental studies in this work were generalized by the following formulas:

when $$\frac{\mu_{ж}^{2}}{\rho_{ж}\sigma_{ж} d_{c}} < 0{,}005$$

$$\frac{d}{d_c} = C_0\left(\frac{\rho_г v^2 d_c}{\sigma_ж}\right)^{-0,45}; \qquad (14)$$

when $0,005 < \dfrac{\mu_ж^2}{\rho_ж \sigma_ж d_c} < 0,5$

$$\frac{d}{d_c} = \left[C_0 + 1,24\left(\frac{\mu_ж^2}{\rho_ж \sigma_ж d_c}\right)^{0,02}\right]\left(\frac{\rho_г v^2 d_c}{\sigma_ж}\right)^{-0,45}; \qquad (15)$$

when

$$\frac{\mu_ж^2}{\rho_ж \sigma_ж d_c} > 0,5$$

$$\frac{d}{d_c} = \left[C_0 + 0,94\left(\frac{\mu_ж^2}{\rho_ж \sigma_ж d_c}\right)^{0,28}\right]\left(\frac{\rho_г v^2 d_c}{\sigma_ж}\right)^{-0,45}. \qquad (16)$$

In these formulas $\mu_ж$, $\rho_ж$ and $\sigma_ж$ are the dynamic viscosity, the density, and the surface tension of the tested liquids.

With the aid of empirical formulas an experimental material was generalized in works [11], and [16] during the analysis of air atomization of liquids. With air-mechanical fuel atomization the results of our experiments were represented by this criterial dependence:

$$\frac{d}{d_c} = 1,21\left(\frac{\mu_т^2}{\rho_т \sigma_т d_c}\right)^{0,0186}\left(\frac{\rho_в v^2 d_c}{\sigma_т}\right)^{-0,334}, \qquad (17)$$

where $v = v_в - v_т$.

Furthermore, we made an attempt to obtain a simplified generalized formula, which would be valid (within the limits of our experiments) both with mechanical, and with air-mechanical fuel atomization.

Analysis of the derived formulas indicates that with mechanical and air-mechanical fuel atomization the basic parameter of the process affecting the fineness of atomization is the velocity of discharge of the fuel (or excess pressure). With the increase in the discharge velocity the diameter of the drops is decreased.

In connection with this we searched for the dependence

$$\frac{d}{d_e}=f(v_{cm}), \qquad (18)$$

where v_{cm} – the discharge velocity of the fuel-air mixture

$$v_{cm}=\frac{v_T+kv_B}{1+k} \qquad (19)$$

Here $k=\frac{G_B}{G_T}$ – the ratio of the flow rate of atomizing air to the flow rate of fuel.

Using the data, obtained experimentally in mechanical and air-mechanical fuel atomization, we obtained an equation of the type

$$\frac{d}{d_e}=0{,}817\left(\frac{V_T+kV_B}{1+k}\right)^{-0{,}585} \qquad (20)$$

which is valid both for air-mechanical, as well as mechanical fuel atomization. In mechanical atomization equation (20) assumes the form

$$\frac{d}{d_e}=0{,}817v_T^{-0{,}585}. \qquad (21)$$

The velocity exponent of various authors is found within the limits of $\sim n=0.34-0.7$. In our case this superscript is located in the range shown and is equal to 0.585.

We have the closest agreement with works [5, 13]. For example, in work [5] for mechanical fuel atomization $n = 0.54$, and in [13] with air-mechanical atomization $n = 0.6$.

Figure 3 shows the graphic dependence of $\frac{d}{d_e}=f(v)$, where we have plotted the points obtained by experimental and calculation methods using formulas (13), (14) and (17).

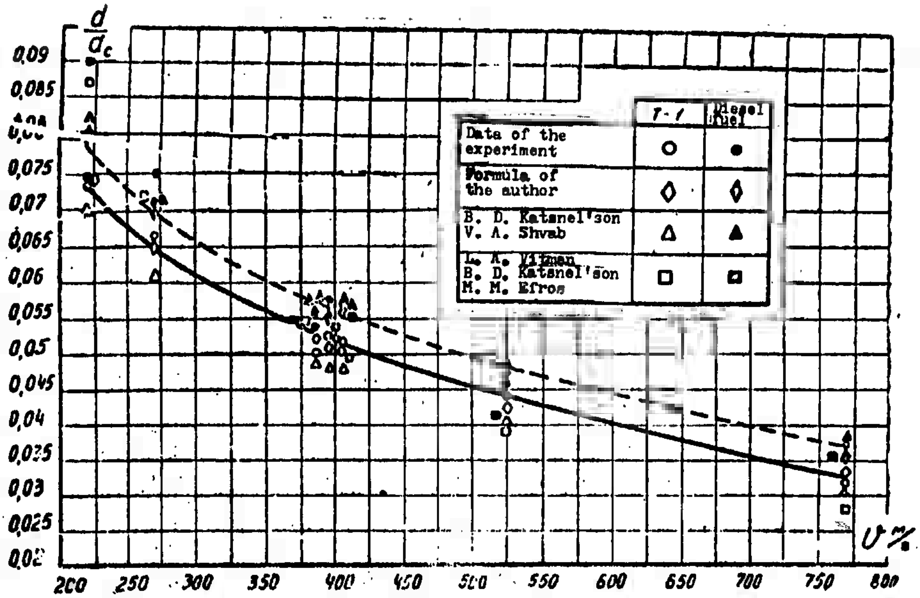


Fig. 3.

One ought to note that formula (14) was selected for reckoning $\frac{d}{d_c}$ on the consideration that it corresponds to the conditions of our experiment with respect to the magnitude of the parameter $\frac{\mu_{ж}^2}{\rho_{ж} \sigma_{ж} d_c}$.

As follows from Fig. 3, the three formulas shown are equivalent and give satisfactory agreement with the experimental data.

Figure 4 gives the experimental points and calculation curve $\frac{d}{d_c} = f(v_{см})$, plotted from formulas (20) and (21).

On this figure we plotted all points obtained experimentally for discharge injectors FR-3 and TVF-1 both with an atomizing air feed and without it to the injectors. The section of the curve over the range v_{CM} = 10-50 m/s corresponds to the operating conditions of the investigated discharge injectors with mechanical fuel atomization.

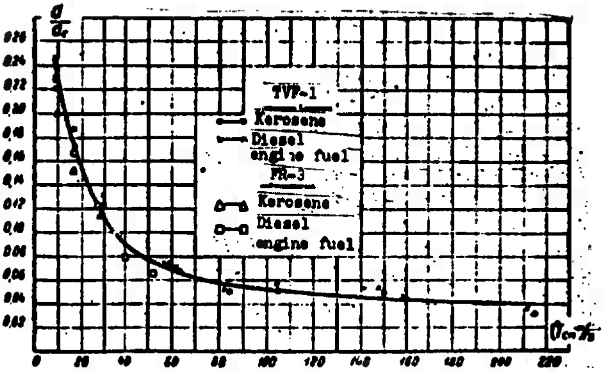


Fig. 4.

As can be seen from Fig. 4, the results of the experiments are adequately described by equations (20) and (21).

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ON THE CALCULATION OF TEMPERATURE FIELD IN
A COOLED TURBINE BLADE WITH
LONGITUDINAL COOLING
CHANNELS

V. I. Lokay and A. V. Sharapov

Introduction

The calculation of temperatures T under the body of a blade with internal cooling requires, in general, the solution to the spatial problem of thermal conductivity with variable boundary conditions both from the side of the gas $T_г^*$, $\alpha_г$, as well as from the side of the coolant $T_в^*$, $\alpha_в$.

For bodies of complex form an accurate analytical solution to such a problem has still not been found. Therefore, in practice, in an application to cooled blades, this is solved either by similarity methods (electrothermal, hydrothermal), or by approximation analytical methods.

In the first case there is required special, very complex and expensive instrumentation (for example, grid integrators); in the second (for example, when using numerical methods: finite differences, "relaxations" [1], and others) to get good accuracy, lengthy calculations are necessary.

Calculations are considerably simplified, if we divide the complex spatial problem into two independent ones: determination of $\bar{T}=\bar{T}(x)$ when $T=T(y,z)=T_{cp}=\text{const}$; a search for $T=T(y,z)$ in several cross sections over the height of the blade with $\frac{dT}{dx}=0$ within the limits of each of the cross sections.

Numerous calculations, their juxtaposition with more accurate solutions and experimental data indicate that with such a method the necessary accuracy is obtained only for points of a blade body, situated in the interval $0{,}25<\bar{x}<1$, where $\bar{x}=\frac{x}{l}$; l – the height of the blade.

In lower cross sections $(\bar{x}<0{,}25)$ because of the heat removal into the locking part of the blades greater longitudinal gradients $\frac{dT}{dx}$ are obtained, and the degree of accuracy proves to be inadequate. But it is just these cross sections that require greatest accuracy in the calculations, inasmuch as they are the most heavily loaded.

For thin-walled blades, or blades with a large number of longitudinal cooling passages, with respect to the type in Fig. 1a, in the search for a solution to $\bar{T}\approx T(x)$, it is possible to make use of the results of works [2, 3].

The distribution of the temperatures in the cross section of a blade being cooled is subject [1] to the Laplace equation

$$\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0 \tag{1}$$

with nonuniform boundary conditions with respect to contour L from the gas side

$$\lambda\frac{dT}{dn}dL=\alpha_r(T_r^*-T)dL; \tag{2}$$

and from the air side

$$-\lambda\frac{dT}{dn}dL=\alpha_u(T-T_u^*)dL. \tag{3}$$

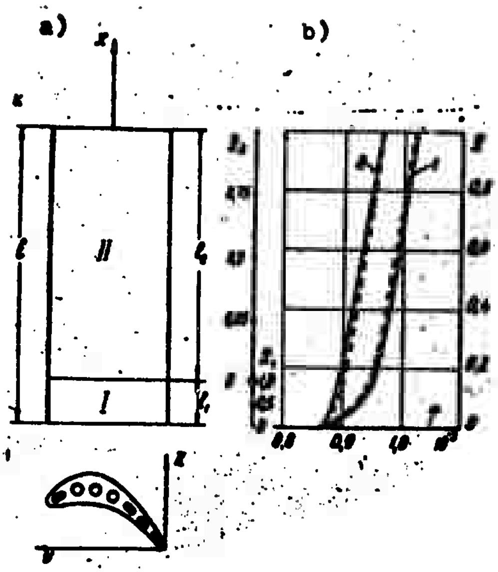

Fig. 1. Determination of average cross section temperatures along the height of the blade: a) a calculation scheme; b) the results of calculation; - - - - from works [2, 3]; ——— - according to the proposed method; $1-\bar{v}_a = f(x)$; $2-\bar{v}_b = f(x)$.

The solution of the system (1)-(3) in general is again reduced to lengthy calculations. Furthermore, in the division of a curvilinear contour into sections the latter is obtained stepwise, with "distorted" boundary conditions.

For this very reason frequently also in the transverse cross section of the blades in preliminary calculations they are limited to the analysis of one-dimensional problems [4], splitting the airfoil into sections, the configuration of which (a plate, a cylinder, a wedge, etc.) allows the use of known "accurate" solutions.

In this work we describe a composite method for the calculation of the temperature field of blades with longitudinal cooling passages. Its basic advantage is a small labor input with completely acceptable accuracy for the estimated variant calculations.

The distribution of temperatures along the height of the blade is executed analytically, in transverse cross sections - by the method of electrothermic analogy with a standard device of type EGDA-9.60.

A method provides account of the longitudinal gradients of temperature when searching for $T = T(y, z)$. The latter is attained by a special correction of the boundary conditions on sections of the blade profile.

Calculation of the Average Cross Section Temperature Along the Height of the Blade

A calculation diagram has been given in Fig. 1. Distribution of the average cross section temperature by height of a blade with longitudinal cooling channels under steadied conditions is approximately described by the system of two equations [2]

$$\lambda f \frac{d^2 T}{dx^2} + \alpha_{\text{г}} u_{\text{г}} l (T^*_{\text{г}} - T) - \alpha_{\text{в}} u_{\text{в}} l (T - T^*_{\text{в}}) = 0; \tag{4}$$

$$G'_{\text{в}} c_{p\text{в}} \frac{dT^*_{\text{в}}}{dx} - \alpha_{\text{в}} u_{\text{в}} l (T - T^*_{\text{в}}) = 0, \tag{5}$$

where λ – the coefficient of thermal conductivity; f – the area of transverse cross section; $\alpha_{\text{г}}$, $\alpha_{\text{в}}$ – the average values of the coefficient of heat transfer from the gas to a blade and from a blade to the air; $u_{\text{г}}$, $u_{\text{в}}$ – the perimeters of a blade profile, flowed around by gas and air; $T^*_{\text{г}}$, $T^*_{\text{в}}$ – the average temperatures of stagnation of the gas and air; $G'_{\text{в}}$ – the hourly flow rate of cooling air; $c_{p\text{в}}$ – the heat capacity of the air. The solution of the system (4)-(5) is shown in work [2].

The deficiency of this solution is that it requires extremely high calculation accuracy for all its components, since the result, which is expressed in the form of the difference in the big numbers, is by 2-3 orders less than the latter.

For our goal, which consists in obtaining the formulas which correct the boundary conditions in various cross sections along the height, without a great disadvantage for accuracy, the solution can

be simplified. In accordance with the actual character of the distribution of the temperature of a blade along its height, the latter can be divided into 2 sections (Fig. 1a): I – the length l_1, where there is essentially heat removal into the blade root ($0 < \bar{x} \leqslant 0{,}15-0{,}25$); II – the length l_2, where the heat transfer along the body of the blade can be disregarded ($\lambda = 0$).

Assuming for section I $(\bar{T} - \bar{T}_в^*) = \mathrm{const}$ and introducing designation

$$\bar{x}_1 = \frac{x_1}{l_1},\quad \bar{T} = \frac{T}{1000°K},\quad \bar{T}_r^* = \frac{T_r^*}{1000°K},\quad \bar{T}_в^* = \frac{T_в^*}{1000°K},$$
$$\tau = \bar{T}_r^* - \bar{T} - \frac{\alpha_в u_в}{\alpha_r u_r}(\bar{T} - \bar{T}_в^*), \tag{6}$$

in place of (4) we get

$$\frac{d^2\tau}{d\bar{x}_1^2} - \kappa^2\tau = 0; \tag{7}$$

here
$$\kappa^2 = \frac{\alpha_r u_r}{\lambda_к f_к} l_1^2,$$

where the subscript "к" indicates that the quantity refers to a root cross section.

Using a boundary condition with $\bar{x}_1 = 0$, $\bar{T} = \bar{T}_к$,

$$\tau = \tau_к = \bar{T}_r^* - \bar{T}_к - \frac{\alpha_в u_в}{\alpha_r u_r}(\bar{T}_к - \bar{T}_в^*),$$

solution (7) can be written in the form

$$\tau = \tau_к \frac{\mathrm{ch}[\kappa(\bar{x}_1 - B)]}{\mathrm{ch}(\kappa B)}. \tag{8}$$

Arbitrary constant B will be found from the second boundary condition at the junction of the I and II sections. At $\bar{x}_1 = 1$ and $\bar{x}_2 = 0$ it is evident that

$$\left(\frac{d\bar{T}}{d\bar{x}_1}\right)_{\bar{x}_1=1} = \left(\frac{d\bar{T}}{d\bar{x}_2}\right)_{\bar{x}_2=0}. \tag{9}$$

Assuming for section II that $\lambda \approx 0$, exclusing from system (4)-(5) the variable $T_в^*$, and introducing the designations

$$\bar{T} - \bar{T}_г^* = \theta, \quad \alpha_г u_г = \kappa_1, \quad \alpha_в u_в = \kappa_2,$$
$$\frac{\kappa_1 \kappa_2}{\kappa_1 + \kappa_2} \cdot \frac{l_2}{G'_в c_{pв}} = m,$$

we will obtain

$$\frac{d\theta}{d\bar{x}_2} + m\theta = 0. \tag{10}$$

The solution of the equation (10), describing the distribution of the temperature of the blade along section II, has the form

$$\theta = \theta_0 e^{-m\bar{x}_2}, \tag{11}$$

where $\theta_0 = \theta_{\bar{x}_2=0}$.

It is evident that

$$\left(\frac{d\bar{T}}{d\bar{x}_2}\right)_{\bar{x}_2=0} = -m\theta_0. \tag{12}$$

Having taken a derivative from equation (8) and taking into account that in (6) $(\bar{T} - \bar{T}_в^*)_{\bar{x}_1} = \mathrm{const}$, we obtain

$$\left(\frac{d\bar{T}}{d\bar{x}_1}\right)_{\bar{x}_1=1} = \kappa\tau_\kappa \frac{\mathrm{sh}\,[\kappa(1-B)]}{\mathrm{ch}\,(\kappa B)}. \tag{13}$$

Now, with account of equality (9), after small transforms we find

$$B=\frac{1}{2\kappa}\ln\left\{\frac{e^{\kappa}\left[\varepsilon+(a+1)e^{\kappa}\right]}{a-ae^{\kappa}-1}\right\}, \tag{14}$$

where $\frac{\kappa}{m}=a;\quad \frac{\kappa_2}{\kappa_1\tau_{\kappa}}(\overline{T}_{\text{к}}-\overline{T}^{*}_{\text{вх}})=\varepsilon.$

Let us note further, that on the basis of designation (6)

$$\Theta_0=\frac{\kappa_2}{\kappa_1}(\overline{T}_{\text{к}}-\overline{T}^{*}_{\text{вх}})-\tau_{\kappa}\frac{\operatorname{ch}[\kappa(1-B)]}{\operatorname{ch}(\kappa B)}. \tag{15}$$

Thus, the distribution of the temperature along section I is determined by expression (8) with account of (14); along section II it is determined by (11) with account of (15).

Figure 1b by continuous lines shows the results of the calculation of the temperatures of a blade at various flow rates of cooling air. The place of the junction of sections was selected at $\overline{x}=0.15$. Basic data corresponded to parameters of contemporary gas turbine [GTD] (ГТД):

$$T^{*}_{\text{г}}=1210^{\circ}\text{K},\quad \alpha_{\text{г.ср}}=1293\ \text{W/m}^2\ \text{deg}.$$

At $\overline{G}_{\text{в}}=2$ and 3% $\alpha_{\text{в.ср}}=1435$ and 1970 W/m^2 deg.

There by the dotted line shows the results of calculations from a refined method[1] [2]. At present the convergence is quite good.

Account of the Longitudinal Heat Flows in the Body of a Blade During the Calculation of Temperatures in a Transverse Cross Section by the Electrothermal Unit Method

As was already indicated, the effect of the longitudinal heat flows on the distribution of temperature in the transverse cross

[1]Calculations conducted engineer G. M. Sal'nikov.

sections of a blade during calculations on analogues from an electroconductive paper can be taken into account by the appropriate correction of the boundary conditions.

Let us examine the temperature state of shaded elements in Fig 2.

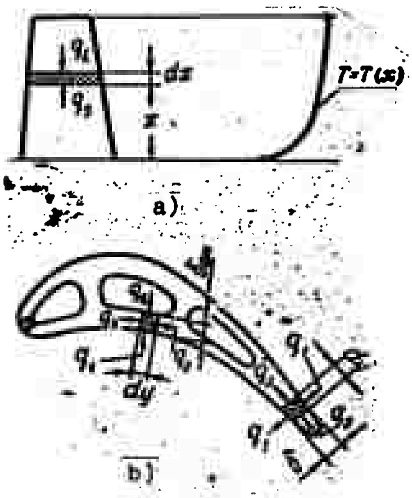

Fig. 2. Taking into consideration of longitudinal flows of heat during the calculation of temperatures in a transverse cross section by the ETA method: a) the change in a temperature over the height of the blade; b) the transverse cross section of a blade at a distance x from the root.

The Trailing Edge

In a steady mode from the equation of thermal balance of the element of the trailing edge

$$2q_1 + q_2 - q_3 + q_4 - q_5 = 0, \tag{16}$$

where

$$q_1 = 2\alpha_r (T_r^* - T)\,dx\,dy;$$
$$q_2 - q_3 = d\left(\lambda\delta \frac{dT}{dy}\right) dx;$$
$$q_4 - q_5 = d\left(\lambda\delta \frac{dT}{dx}\right) dy.$$

For an edge of constant thickness, for example, assume λ = const, we obtain

$$\frac{d^2T}{dy^2}+\frac{2\alpha_r}{\lambda\delta}(T_r^*-T)+\frac{d^2T}{dx^2}=0. \tag{17}$$

We give to this equation the form, which it would have had in the absence of longitudinal flows of heat

$$\frac{d^2T}{dy^2}+\frac{2\alpha_{r.np}}{\lambda\delta}(T_r^*-T)=0, \tag{18}$$

where $\alpha_{r,np}$ — corrected to longitudinal heat flows is the quantity of the arbitrary coefficient of heat transfer.

Comparing equations (17) and (18) and passing over to dimensionless values according to the designations in (6), finally we will receive

$$\frac{L_{r.np}}{L_r}=1+\frac{\dfrac{d^2\bar{T}}{d\bar{x}^2}}{\dfrac{2\alpha_r l^2}{\lambda\delta}(\bar{T}_r^*-\bar{T}_{cp}(x))}. \tag{19}$$

Here $\frac{d^2\bar{T}}{d\bar{x}^2}$ and $\bar{T}_{cp}(x)$ are determined from equations (8) or (11) depending on the location of cross sections along the height of a blade and are considered to be constant for a whole cross section. Analogically, there will also be found corrected quantities $\alpha_{r.np}$ for other sections of a cross section.

Convex and Concave Parts of the Airfoil

From the equation of the thermal balance of an element, by directing the moving coordinate y along the enclosure of the airfoil, it is possible to write:

$$q_1+q_2-q_3+q_4-q_5-q_6=0, \tag{20}$$

where

$$q_1 = \alpha_г (T^*_г - T)\,dxdy;$$
$$q_2 - q_3 = d\left(\lambda\delta_{ст}\frac{dT}{dy}\right)dx;$$
$$q_4 - q_5 = d\left(\lambda\delta_{ст}\frac{dT}{dx}\right)dy;$$
$$q_6 = \alpha_в \left(\frac{u_в}{u_г}\right)_{вн}(T - T^*_в)\,dxdy;$$
$$\left(\frac{u_в}{u_г}\right)_{вн} = \frac{du_в}{du_г} = \frac{du_в}{dy} = p,$$

here p – the relationship of the element surfaces, which come into contact with air and gas.

From equation (20) with $\delta_{ст}=\mathrm{const}$ (the average thickness of a wall) and $\lambda = \mathrm{const}$ it is easy to obtain

$$\frac{d^2T}{dy^2} + \frac{\alpha_г}{\lambda\delta_{ст}}(T^*_г - T) - \frac{\alpha_в}{\lambda\delta_{ст}}\,p\,(T - T^*_в) + \frac{d^2T}{dx^2} = 0. \qquad (21)$$

Considering this, analogically to what was done for trailing edge, and introducing the designation

$$a = \frac{1 + p\dfrac{\alpha_в \bar{T}^*_в}{\alpha_г \bar{T}^*_г}}{1 + p\dfrac{\alpha_в}{\alpha_г}}, \qquad (22)$$

after transforms we obtain

$$\frac{\alpha_{г.эф}}{\alpha_г} = 1 + \frac{\dfrac{d^2\bar{T}}{d\bar{x}^2}}{\dfrac{\alpha_г l^2}{\lambda\delta_{ст}}\left(1 + p\dfrac{\alpha_в}{\alpha_г}\right)\left(a\bar{T}^*_г - \bar{T}_{ср}(x)\right)}. \qquad (23)$$

Here $\dfrac{d^2\bar{T}}{d\bar{x}^2}$ and $\bar{T}_{ср}(x)$ are defined just as in formula (19).

The Leading Edge

Depending on the construction formulation of leading edges the correction α_r may be carried out using formula (19) or (23).

Results of Calculations

In the upper part of Fig. 3 the continuous line indicates the average airfoils by sections (leading and trailing edges, concave and convex parts) the heat transfer coefficient (α_r) from the gas to the surface of a blade. Quantities α_r have been calculated from known criterial equations for parameters in the root cross section of a lattice.

A dotted line indicates the results of the calculation of the corrected boundary conditions (with account of the effect of the longitudinal heat flows) using formulas (19), (23). The quantity $\frac{d^2T}{dx^2}$ in this instance was calculated on the basis of the refined analytical solution, given in work [2].

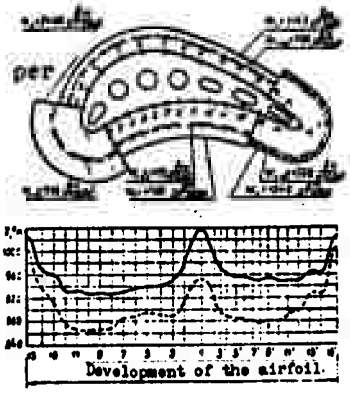

Fig. 3. Boundary conditions and the temperature field over the enclosure of an airfoil in the root cross section of a blade: ——— without correction α_r; ---- with correction α_r calculation $\frac{d^2T}{dx^2}$ from work [2]; -·-·- with correction α_r (calculation $\frac{d^2T}{dx^2}$ from the proposed method).

Designations: $\frac{}{}$ = Watt; = m^2 deg.

A dot-dash line shows the results of the analogous calculations from formulas (19), (23), but $\frac{d^2T}{dx^2}$ was determined on basis of approximation solutions (8), (11).

Calculations indicate that for a uniform temperature field of gas in the upper half of a blade the correction of the quantity for the effect of longitudinal heat flows proves to be so small (1-3%), that it cannot be taken into account; for a nonuniform temperature field of gas the correction of boundary conditions should be made over the entire altitude. The aforesaid is visually confirmed by Fig. 3, where in the lower part is indicated the distribution of temperatures on the enclosure of a blade airfoil in a uniform temperature field of gas.

From Fig. 3 it follows that:

1. During the determination of temperatures by the electrothermal analogy method one should of necessity make a correction to the boundary conditions for the longitudinal heat flow.

2. In the search for the correction of quantities a_r good accuracy is attained, if we use the approximation method for the calculation of $T(x)$, given in this work.

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TURBULENCE IN THE ZONE OF INTERPLAY OF JETSTREAMS WITH THE FLOW

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Yu. S. Alekseyev and G. M. Shalayev

The necessity for knowledge of the turbulent structure of flow in solving problems connected with the movement of a liquid or of a gas has in recent years required ever greater importance. Many of the issues of contemporary gas dynamics, heat and mass exchange cannot be adequately decided without a knowledge of the laws of turbulent motion, without an insight into the mechanism of the phenomenon. The turbulent characteristics of a flow to a considerable degree determine the occurrences of the processes in the combustion chambers of an engine – carburetion, ignition of the fuel, stability of the combustion processes, propagation of the flame, intensity of burn-up of the fuel, heat interchange between the combustion products and the chamber wall, etc.

Theoretical methods of determining the characteristics of turbulence today do not exist; these issues are solved, mainly, by the experimental method. Measurements connected with the detection of the turbulent features of a flow refer to the most complex aerodynamic measurements.

Of great interest is research on a turbulent structure in the zone of flow disruption, especially ~~during~~ with flow around screens, because this will allow us to clarify still-not-satisfactorily explained issues, connected with the mechanism stabilization of the flame. Especially interesting in this respect is the little-studied issue of turbulence in jetstreams [1] and in the zone of interplay of single circular [2], i.e., series of circular, plane [3] and fan jetstreams with deflecting flow. A knowledge of the structure in the blow zone of the jetstreams into a deflecting flow will be ~~a~~ very useful not only in the solution of problems connected with the organization of the working process in the combustion chamber, but also in all cases where intersecting flows are used.

For the measurement of turbulent characteristics in air flows several methods exist. The most effective of these is considered to be the method using the cooling-power anemometer, the sensitive cell of which is a fine metal filament, heated by an electrical current. The filament is included in the circuit of a bridge. The measurement of the characteristics of the current, passing through the filament during flow around it by an air flow, allows us to judge the character of this flow.

For the measurement of turbulent fluctuations of the rate two methods are used: the method of "constant current" ("of dependent resistance") and the method of "constant temperature" ("of constant resistance " filament). The second method possesses definite advantages in comparison with the first. It allows the conducting of measurements in flows with high intensity of turbulence with considerably greater accuracy and in a wider range of frequencies of fluctuations [5].

A feature of a flow in a blast into the deflecting flow of transverse jetstreams or curtains, small in size, but with high velocities is the presence of large velocity gradients, and a wide spectrum of the frequencies and of scales. Therefore to investigate turbulence in intersecting flows the method of "constant temperature" filament has been selected.

For this goal the cooling-power anemometer was created. As the basis of this is the fundamental diagram of the cooling-power anemometer TA-1, developed by G. F. Apollonov and G. V. Smirnov in the laboratory of aerodynamics of the LPI named after Kalinin and modified in the laboratory of turbulent combustion of IKhKIG of the CO (Academy of Sciences) (city of Novosibirsk) by I. I. Kuznetsov. The sensor of the instrument is a tungsten filament with a diameter of 0.008 mm and a length of 4 mm. With the shown sensor sizes the diagram ensures the exception of the thermal mass inertia of the filament over the range of frequencies up to 50 kHz. The effective value of the alternating component of the voltage at the output of the instrument was measured with the aid of an F506 millivoltammeter. For the measurement of the constant component of voltage a composite instrument the Ts57 is used. Observation of the change in the alternating component of voltage was conducted with the aid of the E0-4 electronic oscillograph.

For calibrating a cooling-power anemometer and conducting measurements the composite sensor, schematically shown in Fig. 1, was created. It consists of filament 1 – the sensor of the cooling-power anemometer and of the orientable pneumometric tube. With the aid of opening 2 and 5 total and static pressure is measured, while opening 4 serves to orient the tube in the direction of the velocity at the point, where the measurement is made. The sensor is also provided with chromel-konel thermoelectric couple 3, the joint of which is located in the immediate vicinity of the filament.

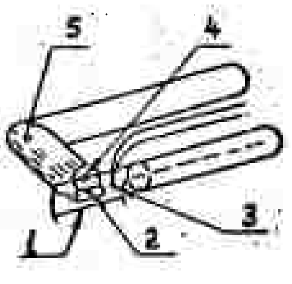

Fig. 1. The diagram of a composite sensor.

The intensity of the turbulence is a ratio

$$\varepsilon = \frac{\overline{W'}}{\overline{W}},$$

where W' — the root-mean-square value of the pulsation component of the flow rate; $\overline{W}$ — the time-averaged value of the flow rate at a given point.

The pulsation component of the flow rate at a given point is determined in the following manner.

The composite sensor is placed into the air flow. By means of the change in the flow rate the dependence $\overline{U} = f(\overline{W})$ (curve 1, Fig. 2) is established, where $\overline{U}$ — the average value of the voltage at the outlet of the cooling-power anemometer; $\overline{W}$ — the average flow rate, measured with the aid of a pneumometric tube. By graphic differentiation of curve 1 the dependence of the sensitivity of the cooling-power anemometer $k = \frac{\partial \overline{U}}{\partial \overline{W}}$ upon the average flow rate $\overline{W}$ can be found (curve 2, Fig. 2). The root-mean-square value of the pulsation component of flow rate at any point is determined by the relationship

$$\overline{W'} = \frac{\overline{U'}}{k},$$

where $\overline{U'}$ — the effective value of the pulsation component of voltage at the outlet of the cooling-power-anemometer; k — the sensitivity of the cooling-power anemometer (it is assumed from the graph that $k = f(\overline{W})$ for a measured average velocity at the given point $\overline{W}$).

To check the operation of the cooling-power anemometer measurement of the intensity of turbulence was conducted after orthogonal lattices. Lattices of three sizes were taken: No. 1 – 8 × 4, No. No. 2 – 12 × 14, No. 3 – 16 × 8 (the first numerals indicate the spacing of the lattice into mm, the second – the diameter of a rod, also in mm).

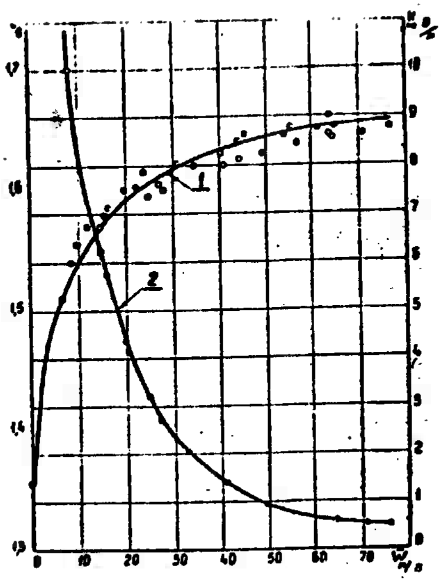

Fig. 2. Calibration curves of a cooling-power anemometer.

Figure 3a depicts the results of the measurement of the intensity of turbulence over the radius of the flow, effusing from a duct (D = 140 mm) at a distance x = 20 mm from the lattice. The intensity of the turbulence remains approximately constant in the flow nucleus and sharply increases in the turbulent layer at the jetstream boundary. The character of the curve corresponds to the results of measurements of ε, given in work [6].

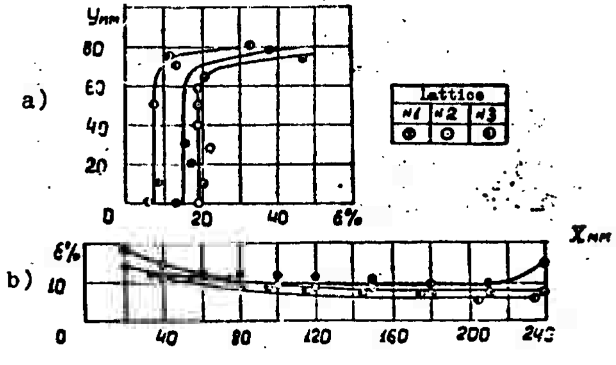

Fig. 3. The intensity of turbulence behind lattices.

Figure 3b shows the change in the intensity of turbulence behind lattices along the axis of air flow. In moving away from a lattice the intensity of the turbulence falls somewhat and only at some distance from it increases, which agrees with the data [4, 7, 8, 9, 10]. Absolute values of ε also rather nearly correspond to the measurements of other investigators (7-15% at a distance of 100 mm from a lattice).

The results of measurements of the intensity of turbulence in the air flow, flowing around a cone with a diameter of 60 mm, are presented in Fig. 4. The greatest values of ε were measured in a turbulent layer directly after the edges of the cone, in the area where disruption of the flow occurs. The intensity of turbulence in the zone of reverse currents in the wake behind the cone comprises ε = 40-50%. These results adequately agree with the measurements of the intensity of turbulence behind the trough-shaped flame stabilizer with a width of 60 mm [4].

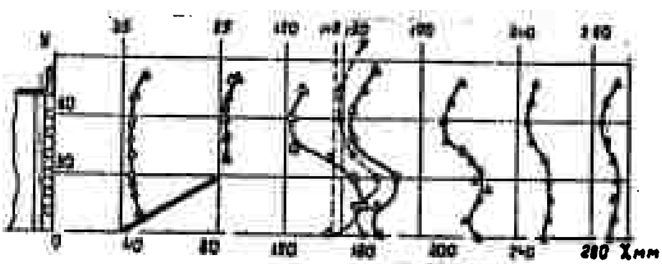

Fig. 4. The intensity of turbulence in the flow around a cone with an angle of 60° at the tip: —·— experimental data; ---- the given works [4].

An analysis of the intensity of turbulence in the zone of interplay of paired plane jetstreams with a deflecting flow was made in a plane chamber with dimensions 350 × 60 mm, one of the lateral walls of which could move within the limits of the working section

560 mm in the length. Along the axis of the chamber an exploring tube with a width of 60 mm and a height of 20 mm was set up. To a spout of the tube were fastened interchangeable fittings, with the aid of which it is possible to change the size of slit b_0 and the angle of outlet of the jetstreams β_0. Measurements were conducted at constant values of the deflecting flow velocity $W = 67$ m/s, the temperature of the deflecting flow $T^*_W = 333°K$, and of the blown-on jetstreams $T^*_V = 283°K$ in the vertical plane of symmetry of the chamber. The necessary ratio of the high-velocity pressure heads of the jetstreams and of the flow $\bar{q}_V = \frac{\rho_V V^2}{\rho_W W^2}$, was established by changing the air pressure in front of the slit.

Figures 5, 6, and 7 show the results of analysis, where numerals 1 and 2 designate the lines of maximum velocities (the trajectory of the jetstream), 3-4 designate the boundary of the zones of reverse currents, and 5 and 6 indicate points of experimentally measured intensity of turbulence.

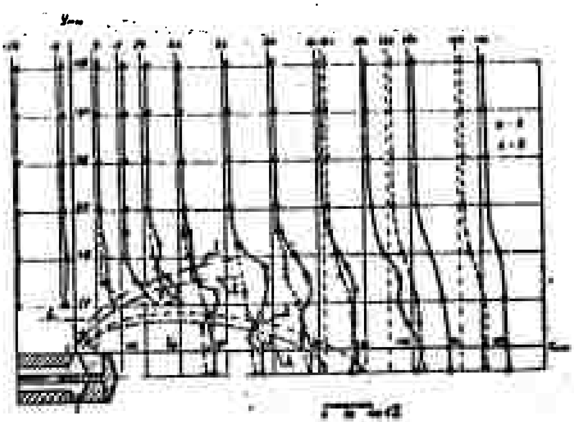

Fig. 5. The change in the intensity of turbulence in the interplay of jetstreams with a flow ($\bar{q}_V = 20$, $\beta_0 = 90°$): 1, 3, 5 – $b_0 = 1.0$ mm; 2, 4, 6 – $b_0 = 0.5$ mm.

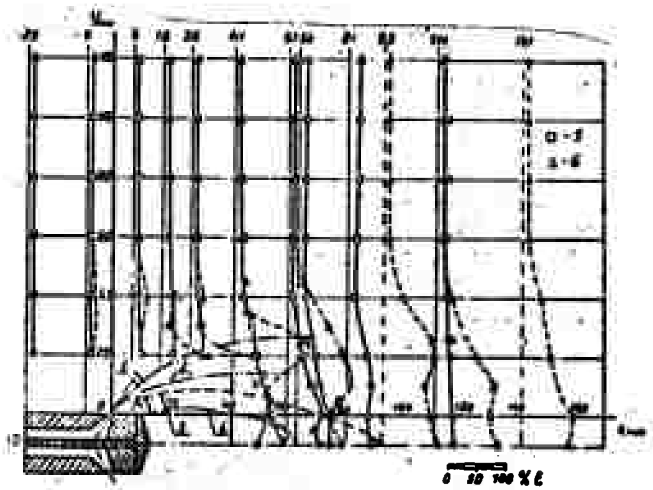

Fig. 6. The change in the intensity of turbulence in the interplay of jetstreams with a flow ($b_0 = 2$ мм, $\beta_0 = 60°$): 1, 3, 5 – $\bar{q}_V = 20$; 2, 4, 6 – $\bar{q}_V = 5$.

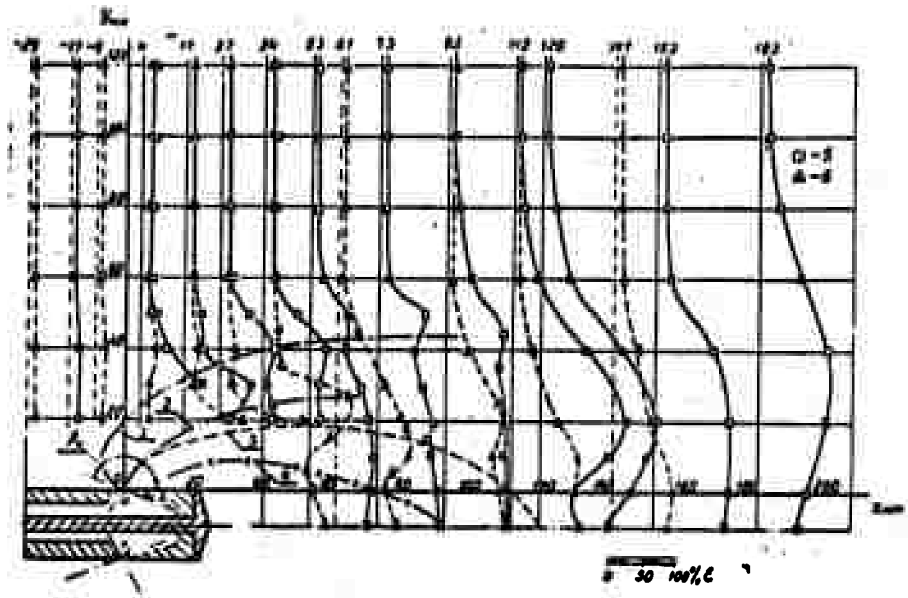

Fig. 7. The change in the intensity of turbulence in the interaction of jetstreams with a flow ($\bar{q}_V = 20$, $b_0 = 2$ мм): 1, 3, 5 – $\beta_0 = 150°$; 2, 4, 6 – $\beta_0 = 60°$.

As can be seen from the graphs, the degree of turbulence of an incoming flow amounts to ∿6%. In every cross section an increase in the intensity of turbulence is observed near the lines of maximum velocities, attaining the greatest value in the area between this line and the boundary of the zone of reverse currents. The maximum value of the amount of turbulence (in some modes up to 150%) takes place in the region of the end of the zone of reverse currents.

An analysis of the obtained results indicates that the absolute values of the intensity of turbulence are significantly higher in the zone of interplay of deflecting flow with jetstreams, than with poorly streamlined bodies (Fig. 4) [4, 6].

Figures 5 and 6 show the effect of the size of the slit b_0 and the ratio of the high-velocity pressure heads $\bar{q}_v$ on the intensity of turbulence. With an increase in b_0 (at $\bar{q}_v = \text{const}$) and $\bar{q}_v$ (at $b_0 = \text{const}$) the flow rate of the air increases (the quantity of movement) of the jetstream, which leads to an increase in the width of a turbulent wake, in the slope of the trajectory, and in the size of the zone of circulation (of reverse currents). Together with this, the intensity of turbulence in the area between the line of maximum velocities and the axis of the chamber increases. With a change in b_0 from 0.5 mm to =1.0 mm [sic] the amount of turbulence in the zone of reverse currents is changed from ∿45% to ∿80%, and with a change in $\bar{q}_v$ from 5 to 20 (Fig. 6) – from ∿50% to ∿95%.

With the change in the exit angle β_0 = from 60° to β_0 = 150° (Fig. 7) the jetstream is more deeply interposed into the flow, in consequence of which the size of the zone of reverse currents and the slope of the lines of the maximum velocities increase. An increase in the degree of turbulence begins further from the axis, and its absolute value increases.

From an analysis of the obtained results it is evident that the character of the change in the intensity of turbulence in the interplay of the flow with poorly streamlined bodies and jetstream screens is identical. However, its absolute values in the case of jetstream screens are significantly higher.

The analysis conducted indicates that by means of the change in the parameters of a jetstream it is possible to change the intensity of turbulence in the zone of its interplay with the flow in the desired direction.

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